THE
DRIVING OF MACHINE TOOLS

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THE DRIVING OF MACHINE TOOLS

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PREFACE

THIS book is the outcome of a series of lectures on Machine Tools given by the author at the Royal Technical Institute, Salford, and its object is to give to the readers some simple and rational methods of arranging a drive to give the desired results, and to include such formulæ, rules and data, as are required.

It must be remembered that the problems dealt with are continually being presented to those who have designing to do, and, of course, there are always new students and men coming into this line of work, who, lacking experience, are glad to receive suggestions, even of the simplest character.

While many of the methods and devices are not claimed to be either new or original, they are presented in the hope of being of service, because the more one looks back and considers the things that have been done in practice, and the more old designs are studied, the more one is impressed with the importance of a designer knowing what is settled by experience.

It is a pleasure to acknowledge the many courtesies received from manufacturers, and

the valuable information furnished in the preparation of this book, also my indebtedness to "Machinery," the "American Machinist," and in particular to Mr. Fowler of the "Mechanical Engineer," who very kindly allowed me the use of several illustrations; and lastly, I must not forget a word of thanks to those who assisted in the preparation of the various drawings.

THOS. R. SHAW.

MANCHESTER,
February, 1917.

CONTENTS

CHAPTER I.

INTRODUCTION.

In designing a new machine tool the principal thing to aim at is to ensure that the finished product will successfully fulfil the object for which it is designed. It is not sufficient that the machine will run, any old worn-out machine will do that, but it must be commercially efficient. This does not in any way mean the most complicated design, rather the opposite, because the more simple a machine can be made the more simple it will be to operate, smaller overhead charges will come against it and cheaper labour be necessary to run it; in fact, the machine tool of to-day must meet the conditions of to-day.

When commencing a new design, certain features and dimensions must first be known. The starting-point is usually the capacity of the machine, and after this is determined, the type of machine which will be most suitable for the particular class of work is selected.

Closely allied to this question is the problem of deciding whether a vertical or horizontal spindle machine is best suited. A particular example is to be found in the choice between a lathe or a boring and turning mill. Again, much work could be more quickly and cheaply performed in a vertical drilling

machine than in either a horizontal lathe or boring mill. The question of head-room versus floor-space also should not be overlooked.

Productive capacity—the chief object of the machine—is determined by three principal factors: durability, control, and accessibility.

Durability.—It is reasonable to expect a machine to be sufficiently robust to stand up and do its work, because, naturally, if it is not a durable machine, its ability to produce soon ceases, and it is therefore inefficient. The machine tool builder now has a knowledge of design, variety of material, and resources of construction that were never before available, all tending to the making of more efficient machinery.

The demands of motor-car construction have proved a wonderful stimulus to the metallurgy of special steels. The tremendous duty expected of materials of construction—lightness without sacrifice of toughness or strength—tend to make the requirements of the automobile trade most exacting, and the benefits of scientific research resulting in the extensive use of nickel and chrome-nickel steels is made use of with corresponding advantages by makers of machine tools.

These steel alloys enable the use of parts to give the very highest efficiency and durability. They give the steel wonderful properties, which can be further increased by the processes of heat treatment. A gear can be made from chrome-nickel steel, and after heat-treating the teeth of the gear can be banged over against other teeth without breaking. This cannot possibly be done with a gear of cast iron, or even of soft mild steel, without making it so

unwieldy that its use would be practically impossible. Why is it possible with alloy steel? Because of its physical characteristics giving a tensile strength of 50 to 120 tons per sq. in.

Thus, this great gift of the automobile to the art of machine design is a series of materials possessing in varying degrees the properties of resistance to shock, great tensile strength, and adaptability to heat treatment. To measure results, methods and machines have been developed for shock and alternate stress testing. In a lesser degree there has been improvement in the use of certain copper and aluminium alloys, and the gaining of a better knowledge of their properties and methods of handling.

Hardened gears are also a refinement taken from automobile practice, the advantage of these being that they can run at pitch-line speeds, or surface speeds, considerably higher than the limit for soft steel gears. Bath lubrication on the principal gears and important bearings is also essential in order to provide a sufficient amount of oil that will reduce friction to a minimum and carry away any excess heat generated.

Ball-bearings are largely used on the revolving parts of machine tools, and the result is that the old defect of being unable to convey to the cutting tool sufficient power to use this to its highest capacity has been overcome, and it may be regarded as an axiom that ball-bearings, heat-treated or hardened gears, and bath lubrication must be considered as essentials in the design of machine tools that are required to work to the limit of their capacities.

Then comes the question of control. Control is the essence of machine-tool operation and plays one

of the largest parts in the production of work. The machine itself should do the work, and the operator should use his energy solely in directing it. It should be capable of being started or stopped exactly at the right time, one of the most important functions of the operating mechanism, and for the moving of heavy parts power operation is an essential feature. This, therefore, means a convenient grouping of all operating handles, so that from his working position the operator has control over all movements.

Another point needing consideration is **Accessibility**, a feature often overlooked. All gears and mechanisms for controlling same should be readily accessible for examination or removal without having to dismantle the whole machine before any particular part can be reached.

One of the first problems encountered in the design of a new machine tool is that of laying out the drive. The importance of a properly proportioned drive will easily be recognized, because the use of high-speed steels, and the high pressure under which modern manufacturing is carried on, precludes the use of any but the most modern and efficient drive.

The drive selected may be one of the following different kinds, depending on the conditions surrounding the case in hand: We may make the drive to consist of cone pulleys only; we may use cone pulleys in conjunction with one or more sets of gears; or we may make our drive to consist of gears only, depending on one pulley, which runs at a constant speed, for our power. If the conditions will allow we may use an electric motor, either independently or in connection with suitable gearing.

CHAPTER II.

CONE PULLEYS AND GEARING.

Use of the Cone Pulley.—In many machine tools the power that is required by the machine is transmitted by means of a belt running on the peripheries of two cone pulleys, one on the counter or line shaft, the other on the machine, the provision of these pulleys being to enable a series of different speeds being given to the machine. In conjunction with these cone pulleys a system of gearing is employed, the purpose of this gearing being to allow a high speed of driving belt, which, when reduced through the gearing, allows a high turning effort to be given out for work of large diameter or hard material. The gearing is also sometimes utilized to increase the number of changes of speed which are derived from the use of the cone pulleys, and this is accomplished by arranging some means of throwing out of gear part or the whole of the gearing.

If the cone pulley on the machine has four steps, and the gearing is arranged so that we can disengage part, and then the whole of it, we shall obtain three different arrangements of drive, and in place of having four speeds at our disposal, we shall have twelve speeds available, and then if a two-speed

counter-shaft is used this number is again doubled, giving twenty-four speeds in all, a desirable feature when we have to deal with work of different kinds, both as regards diameter and material.

Importance of Speed Range.—One of the points to which we must pay attention is the provision of a sufficient number of speed changes. The experiments made with high-speed tool steels show that there is a definite relation between the cutting speed and the length of time which a tool will last without regrinding. If the machine be run at too high a speed, the tool will last but a short time before it has to be reground. Should it be run at too low a speed, less work, of course, will be done, although the tool will last a comparatively long time. Somewhere there is a golden mean at which the cost of machining plus the cost of tool dressing is a minimum, and, theoretically, our machine should always be run at that speed. Of course, in handling materials of various grades of hardness, and in the case of lathes and boring and turning mills of largely varying diameters, this would necessitate a very great number of speed changes. If the number of speed changes is limited, it is apparent that the machine cannot always be working at the point of maximum efficiency.

A simple illustration serves to press this point home. Assume a number of shafts of a certain material have to be turned, and it has been found by previous experiment with this material that 50 ft. per min. is the best possible speed to work at, but unfortunately on the lathe on which we have to do this work the nearest spindle speeds available

give cutting speeds of 40 ft. and 60 ft. per min. respectively. If 60 ft. per min. is too high, then we are compelled to use the speed lower—that is, 40 ft. —with the result that we have only $\frac{40}{50}$ of the speed we should have, equivalent to a loss in time of 20 per cent.

This could be illustrated in another way. Assume that it takes 10 hrs. to do the work when using the 40 ft. cutting speed. If we could run at 50 ft., we could do the work in $\frac{40}{50}$ of the time—that is, in 8 hrs. instead of 10—thereby saving a loss of 2 hrs. It should also be borne in mind that this is only the saving due to cutting time, but the actual saving will be considerably greater, as there is to be taken into account the establishment charges, supervision, power, rates and taxes, etc., which come against the machine while on the work.

For many years designers of machine tools were inconvenienced by the want of some logical process for designing cone pulleys and gearing, to give an even graduation between the limits of highest and lowest speeds prescribed by the work to be done, and the only law in existence seemed to be that of trial and error, with the result that the best combination was not discovered. The difficulty of the problem when attacked in this way depends upon the degree of complication in the mechanism to be used, ranging from the simple cone pulley and one-speed counter-shaft to the elaborate combinations of pulleys and gearing used on some of the larger machine tools. It was probably not until the year 1903, when Mr. P. V. Vernon explained a simple method before the members of the Manchester

Association of Engineers, that any serious attempt was made to tackle the problem.

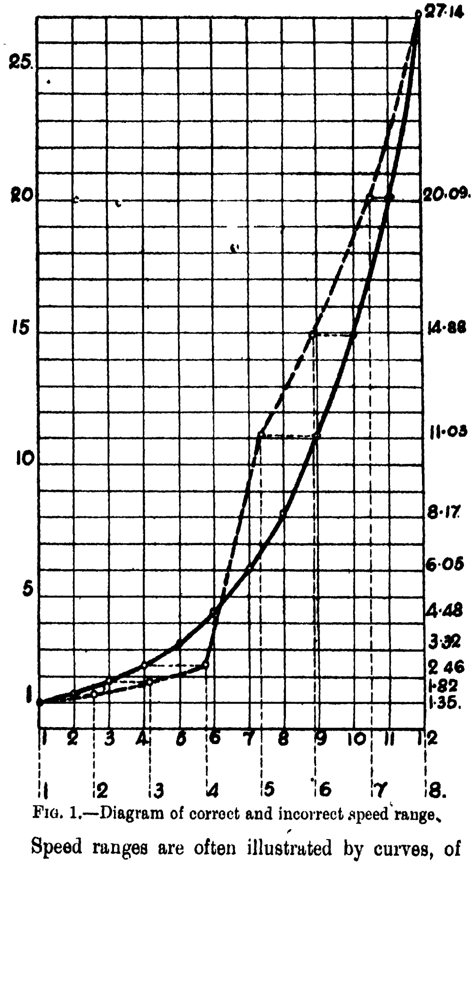


FIG. 1.—Diagram of correct and incorrect speed range.

Speed ranges are often illustrated by curves, of

which a specimen is given in Fig. 1. This diagram contains two curves, and is intended to show the difference between a good and a bad range. The dotted curve represents eight speeds, and it should be noticed that there is too big a gap between each set of four speeds. In the full-line curve is shown a correct range and also the influence of the addition of another set of speeds.

The range of speeds needed is, of course, entirely governed by the type of machine and the work required from it. If, say, for a milling machine, it would depend on the diameter of cutters intended to be used, with an allowance for different cutting speeds for various materials. For a lathe, it would depend on the largest diameter of work proposed to be dealt with down to a small shaft. As an example, if the largest diameter to be machined is 18 ins., and the smallest diameter likely to be dealt with on the machine is $1\frac{1}{2}$ in., this gives a range of speeds of

$$\frac{18}{1\frac{1}{2}} = 12 \text{ to } 1,$$

assuming a uniform cutting speed on all diameters. Then we must take into account the extreme variations in hardness of material, which may sometimes call for a cutting speed of 20 ft. per min., and then again of 60 ft. per min., and the total range then becomes

$$12 \times \tfrac{60}{20} = 36 \text{ to } 1.$$

Efficiency of a Speed Range.—In almost every machine shop there is a great amount of lathe work that does not vary greatly in diameter, and therefore to obtain the maximum efficiency from a lathe for

dealing with this class of work alone, the cone pulley and gearing should be designed in accordance, and not to cover a wide range such as a machine intended to meet the requirements of every one

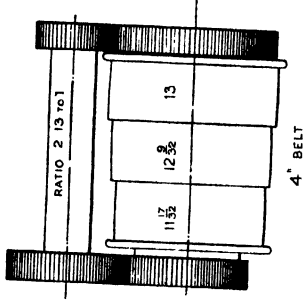


FIG. 2.—Head with low-gear ratio.

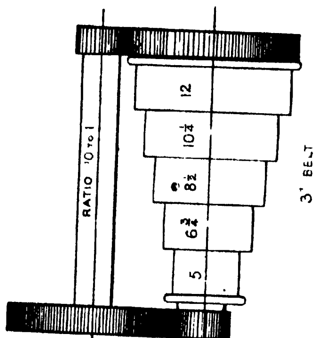


FIG. 3.—Comparative head with higher-gear ratio.

does. As a typical example of this, we will compare the headstock of a lathe specially designed for a low range, such as shafts from 1 to 4 ins. diameter, with one of an ordinary pattern. See Figs. 2 and 3.

If by experience it was found that a certain cutting speed gave the best results, then by the use of this special head it is possible by the use of twelve speeds to have the speeds so graded that none will be lower than 10 per cent of the desired speed over the whole range of diameters, taking these in steps as fine as $\frac{1}{8}$ inch. These speeds are obtained by means of a three-step cone, with diameters varying by small amounts and a back-gear ratio of only 2·13 to 1, with a two-speed counter-shaft. If the ordinary pattern had a five-step cone, ranging from 5 to 12 ins., and a back-gear ratio of 10 to 1, we find the latter is quite unsuitable although with the higher back-gear ratio.

It has been shown that the maximum speed need be but four times faster than the minimum speed, whereas the speeds on the ordinary type have a ratio of 58 to 1, as shown in the accompanying table. If both lathes are speeded so as to cut a 1-in. shaft at a maximum cutting speed of 100 ft. per min., it will be seen from this table that only four of the speeds are adapted to the work, for instance a $3\frac{1}{8}$-in. shaft has to be turned at the same speed of revolution as a 4-in. shaft, with a corresponding loss of time as already pointed out. It will also be noticed that only the speeds without the back gear can be used, so that it will easily be understood which is the more powerful lathe of the two, the back-gear ratio of 10 to 1 being of no use, whereas with the special head the low back gear can be used to full advantage.

TABLE OF SPEEDS OF LATHES TO GIVE AN APPROXIMATELY UNIFORM CUTTING SPEED OF 100 FT. PER MINUTE.

Special Lathe.		Standard Lathe.	
Revs. of Spindle.	Dia. of Work.	Dia. of Work.	Revs. of Spindle.
382	1	1	382
337	$1\frac{1}{8}$	$1\frac{1}{2}$	242
297	$1\frac{1}{4}$	$2\frac{3}{8}$	159
262	$1\frac{3}{8}$	$3\frac{5}{8}$	104
231	$1\frac{5}{8}$		66
204	$1\frac{7}{8}$		
179	$2\frac{1}{8}$		38
158	$2\frac{3}{8}$		24
139	$2\frac{3}{4}$		16
122	$3\frac{1}{8}$		10·4
109	$3\frac{1}{2}$		6·6
96	4		

Geometrical Progression.—After having determined the form which our drive is to take, and the range of speeds necessary to fulfil the requirements, a natural question to ask is: What is the law governing the progression of these speeds? It will be found that the speeds required differ by small increments at the slow speeds, the increment gradually increasing as the speed increases. Speeds laid out in accordance with the rules of geometrical progression fulfil the requirements of the above conditions.

If we multiply a number by a multiplier, then multiply the product by the same multiplier, and continue the operation for a definite number of times, we have in the products obtained a series of numbers which are said to be in geometrical progression.

Thus 1, 2, 4, 8, 16, 32, 64 are in geometrical progression, since each number is equal to the one preceding, multiplied by 2, which is called the ratio.

In getting out a set of speeds the tables on pages 14 and 15 (which were originally published by "Machinery," New York, and extended by the author) are of value, as a *comparative set of speeds* can be taken out direct from the tables. They are especially useful to those who are unfamiliar with the use of logarithms. The tables are self-explanatory, yet may be made clearer by a few examples, and when once understood and mastered, it will be found the essence of simplicity to obtain any desired cone and gear ratio, or ratio of gears for a gear box.

Suppose we have a simple four-step cone and wish to have the entire ratio of speeds obtained about 2½ to 1. We look in the fourth column for the value nearest 2½, and find 2·46. The ratio between two adjoining speeds is given as 1·35. If it was desired to extend the number of speeds by double gearing in the same ratio, the ratio of the gearing required would be in the fifth column—namely 3·32 to 1. Suppose we go still further and introduce another set of gearing, this gear ratio would be found in the ninth column—namely 11·03 to 1, which, it should be noted, is the square of 3·32.

The range of all the twelve speeds thus obtained can be read right across the table, and is 1, 1·35, 1·82, . . . 20·09, 27·14.

Suppose now that we want twenty speeds, the fastest to be thirty times the slowest. From the table under column 20 we should find the ratio between two

NUMBER OF SPINDLE SPEED OR FEED FROM SLOWEST TO FASTEST.

Ratio.	In-crease %.	1.	2.	3.	4.	5.	6.	7.	8.	9.	10.	11.	12.	13.	14.	15.
1·10	10	1	1·10	1·21	1·33	1·46	1·61	1·77	1·95	2·14	2·36	2·59	2·85	3·14	3·45	3·80
1·11	11	1	1·11	1·23	1·37	1·52	1·68	1·87	2·07	2·30	2·55	2·84	3·15	3·49	3·88	4·30
1·12	12	1	1·12	1·25	1·40	1·57	1·76	1·97	2·21	2·47	2·77	3·10	3·47	3· 9	4·36	4·88
1·13	13	1	1·13	1·28	1·44	1·63	1·84	2·08	2·35	2·66	3·00	3·39	3·83	4·33	4·89	5·53
1·14	14	1	1·14	1·30	1·48	1·69	1·92	2·19	2·50	2·85	3·25	3·70	4·22	4·81	5·49	6·26
1·15	15	1	1·15	1·32	1·52	1·75	2·01	2·31	2·66	3·06	3·51	4·04	4·65	5·34	6·15	7·07
1·18	18	1	1·18	1·39	1·64	1·94	2·29	2·70	3·18	3·75	4·43	5·23	6·17	7·28	8·59	10·13
1·20	20	1	1·20	1·44	1·72	2·07	2·48	2·98	3·58	4·30	5·16	6·19	7·43	8·91	10·70	12·84
1·22	22	1	1·22	1·48	1·81	2·21	2·70	3·29	4·02	4·90	5·98	7·29	8·90	10·86	13·25	16·17
1·25	25	1	1·25	1·56	1·95	2·44	3·05	3·81	4·77	5·96	7·45	9·31	11·64	14·55	18·19	22·74
1·30	30	1	1·30	1·69	2·20	2·86	3·71	4·83	6·27	8·16	10·61	13·79	17·92	23·30	30·29	39·37
1·35	35	1	1·35	1·82	2·46	3·32	4·48	6·05	8·17	11·03	14·88	20·09	27·14	36·64	49·47	66·78
1·40	40	1	1·40	1·96	2·74	3·84	5·38	7·53	10·54	14·76	20·66	28·93	40·50	56·69	79·37	111·10
1·45	45	1	1·45	2·10	3·05	4·42	6·41	9·29	13·48	19·54	28·33	41·09	59·57	86·38	125·30	181·5
1·50	50	1	1·50	2·25	3·38	5·06	7·59	11·39	17·09	25·63	38·44	57·67	86·50	129·80	194·6	292·0
1·55	55	1	1·55	2·40	3·72	5·77	8·95	13·87	21·49	33·32	51·64	80·04	124·10	192·2	298·0	462·0
1·60	60	1	1·60	2·56	4·10	6·55	10·49	16·78	26·84	42·95	68·72	110·0	176·0	—	—	—
1·65	65	1	1·65	2·72	4·49	7·41	12·23	20·18	33·30	54·94	90·65	149·6	—	—	—	—
1·70	70	1	1·70	2·89	4·91	8·35	14·20	24·14	41·03	69·75	118·60	—	—	—	—	—
1·75	75	1	1·75	3·06	5·36	9·38	16·41	28·72	50·27	87·96	153·90	—	—	—	—	—
1·80	80	1	1·80	3·24	5·83	10·50	18·90	34·01	61·22	110·20	—	—	—	—	—	—
1·85	85	1	1·85	3·42	6·33	11·71	21·67	40·09	74·17	137·20	—	—	—	—	—	—
1·90	90	1	1·90	3·61	6·86	13·03	24·76	47·05	89·39	169·80	—	—	—	—	—	—
1·95	95	1	1·95	3·80	7·41	14·46	28·20	54·98	107·2	—	—	—	—	—	—	—
2·00	100	1	2·00	4·00	8·00	16·00	32·00	64·00	128·0	—	—	—	—	—	—	—

Table of Geometrical Progression.

Number of Spindle Speed or Feed from Slowest to Fastest.

Ratio.	Increase %.	16.	17.	18.	19.	20.	21.	22.	23.	24.	25.	26.	27.	28.	29.	30.
1·10	10	4·18	4·59	5·05	5·56	6·11	6·72	7·40	8·14	8·95	9·85	10·83	11·91	13·10	14·41	15·86
1·11	11	4·78	5·30	5·88	6·53	7·25	8·05	8·93	9·91	11·00	12·21	13·56	15·05	16·70	18·54	20·58
1·12	12	5·47	6·12	6·86	7·68	8·60	9·64	10·79	12·09	13·54	15·16	16·98	19·02	21·30	23·86	26·72
1·13	13	6·25	7·06	7·98	9 02	10·19	11·51	13·01	14·70	16·61	18·77	21·21	23·97	27·08	30·60	34·58
1·14	14	7·13	8·13	9·27	10·57	12·05	13·73	15·65	17·85	20·35	23·19	26·44	30·14	34·36	39·17	44·66
1·15	15	8·13	9·35	10·75	12·36	14·22	16·35	18·80	21·62	24·86	28·59	32·88	37·81	43·48	50·01	57·51
1·18	18	11·96	14·11	16·65	19·65	23·19	27·36	32·28	38·09	44·95	53·04	62·59	73·86	87·15	102·8	121·3
1·20	20	15·41	18·49	22·19	26·62	31·95	38·34	46·01	55·21	66·26	79·51	95·4	114·4	137·3	164·8	197·8
1·22	22	19·72	24·06	29·36	35·82	43·70	53·31	65·04	79·35	96·81	118·1	144·1	175·8	214·4	261·6	319·2
1·25	25	28·42	35·53	44·41	55·51	69·40	86·74	108·40	135·5	169·3	—	—	—	—	—	—
1·30	30	51·19	66·54	86·50	112·50	146·2	190·0	247·0	321·0	417·5	—	—	—	—	—	—
1·35	35	90·37	121·70	164·29	221·8	299·4	—	—	—	—	—	—	—	—	—	—
1·40	40	155·5	217·8	304·86	426·8	597·5	—	—	—	—	—	—	—	—	—	—
1·45	45	263·3	382·0	554·0	803·0	1164·0	—	—	—	—	—	—	—	—	—	—
1·50	50	437·8	656·7	985·0	—	—	—	—	—	—	—	—	—	—	—	—
1·55	55	716·0	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1·60	60	1152·0	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1·65	65	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1·70	70	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1·75	75	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1·80	80	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1·85	85	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1·90	90	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1·95	95	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
2 00	100	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—

Table of Geometrical Progression.

adjoining speeds would be 1·20, or an increase of 20 per cent.

A little further investigation shows the utility of the table. Set out the range of speeds required and

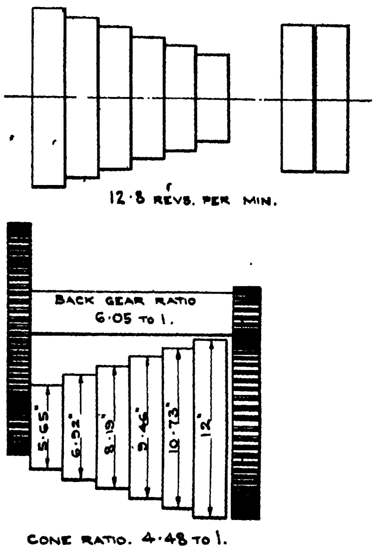


FIG. 4.—Six-step cone pulley.

analyse them, and we shall see some of the combinations obtainable. Taking the same range as before, we have :—

1	Back gear						No gear					
2	2nd gear				1st gear				No gear			
	1	1·35	1·82	2·46	3·32	4·48	6·05	8·17	11·03	14·88	20·09	27·14
3	3rd gear			2nd gear			1st gear			No gear.		
4	2nd counter			1st counter			2nd counter			1st counter		

The speeds bracketed at the top are for a six-speed cone with single-speed counter-shaft. It will be seen that the cone ratio is 4·48 to 1, and the gear ratio 6·05 to 1. The resultant cone and gear are shown in diagram form in Fig. 4.

The second set of speeds bracketed are for a four-speed cone with two changes of gear (see Fig. 5);

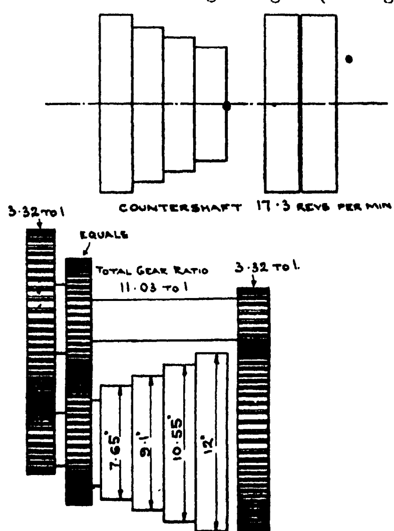


FIG. 5.—Four-step cone pulley and double back gear.

those at the bottom are for a three-speed cone with three changes of gear, as Fig. 6, and two-speed counter-shaft as Fig. 7. In Example 3 the cone ratio is 1·82 to 1, and the gear ratios 2·46, 6·05, and 14·88 to 1. Now assume in the latter case that a two-

speed counter-shaft was used to replace two of the gear changes, we should then have only one back-gear ratio as Example 1, viz. 6·05 to 1, but the cone ratio would remain 1·82 to 1.

Speed of Counter-shaft.—When the cone pulleys

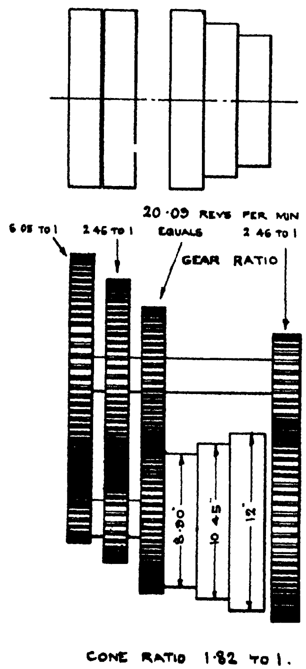


FIG. 6.—Three-step cone pulley and three back gears.

have an *odd* number of steps, the speed of the counter-shaft always equals the speed of the driven

cone on the highest set of speeds when set out in geometrical progression. Referring to our table of figures, and looking at the right-hand end, we find one speed required to be the middle of the first three, 20·09 rev. per min., and for the second speed, the

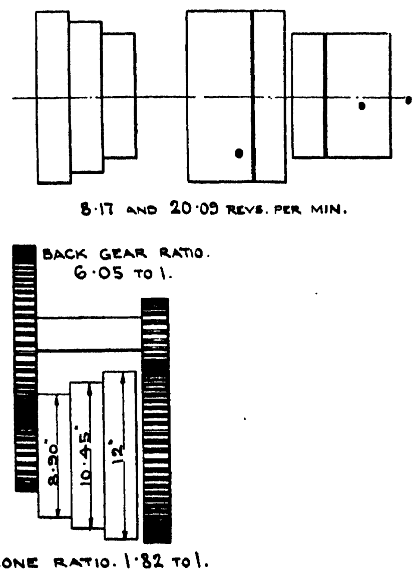


FIG. 7.—Three-step cone pulley and double speed countershaft.

middle of the second set, 8·17 rev. per min. It should be clearly understood that these are all comparative figures, and the ones required can be easily found afterwards. For instance, if the slowest speed required was six instead of one, then all that would

be necessary would be to multiply the whole set of figures by 6.

If the number of steps of the cone pulley is *even*, as in Example 2, then the counter-shaft speed = quickest speed of cone $\times \sqrt{\frac{\text{slowest}}{\text{quickest}}}$ speed of cone.

From our table we get 27·14 as the quickest and 11·03 as the slowest speed, therefore the speed of counter-shaft equals

$$27{\cdot}14\sqrt{\frac{11{\cdot}03}{27{\cdot}14}} = 17{\cdot}3 \text{ revs. per min.}$$

To Obtain the Cone Diameters.—When two cone pulleys are alike on the machine and counter-shaft the *ratio* always equals

$$\frac{\text{Largest diameter}^2}{\text{Smallest diameter}^2} = \frac{D^2}{d^2}.$$

Now taking Example 2 with the four-speed cone, and assume the largest step of the pulley to be 12 ins. diameter, we may then use our ratio from the table 2·46 to 1, and have as follows:

$$\frac{D^2}{d^2} = 2{\cdot}46 \text{ to } 1.$$

$$\therefore d^2 = \frac{12 \times 12}{2{\cdot}46} = 58{\cdot}5,$$

$$\text{and } d = 7{\cdot}65 \text{ ins.}$$

The cone pulley will therefore have steps of 7·65 – 9·1 – 10·55 – 12, the diameters being in *arithmetical progression*, that is, each successive speed is increased by the same amount, in this case being 1·45 in.

Taking Example 4, and again assuming our largest step to be 12 ins. diameter, we have

$$\text{Ratio} = \frac{D^2}{d^2} = 1{\cdot}82 \text{ to } 1$$

$$d^2 = \frac{12 \times 12}{1{\cdot}82} = 79{\cdot}12,$$

and $d = 8{\cdot}90$ ins.

The middle step equals half the sum of the other two steps :

$$= \frac{12 + 8{\cdot}9}{2} = \frac{20{\cdot}9}{2} = 10{\cdot}45 \text{ ins.}$$

The cone pulley will therefore have diameters of 8·9 - 10·45 - 12.

Power required.—For the purpose of illustrating the above method, we have assumed a diameter of cone pulley, but in order to make absolute computations of the required diameters, we should have reliable data on the amount of the pressure or cutting effort at the edge of the tool.

The power required to remove metal depends upon the nature of the cutting tool and the amount of metal removed per minute. Cutting tools may be divided into three classes: (*a*) lathe tool type; (*b*) drills; (*c*) milling cutters.

The lathe tool is used on lathes, boring-mills, planers, shapers, and slotters. Tests show that the power required by a tool of this kind when removing metal depends upon the cutting angle of the tool and the number of cubic inches of metal removed per minute. From observation and data obtained by means of a graphic recording meter, and the use of tools having a cutting angle of about 75° to 80°, the curve shown in Fig. 8 was obtained. The cubic inches of metal removed per minute are found as follows :—

Area of cut (square inches) × cutting speed (feet per minute) × 12.

The area of cut = depth of cut in inches × feed (inches per revolution).

The **horse-power required** to remove metal with the tools ordinarily employed can be expressed by: HP = a constant × cubic inches removed per minute.

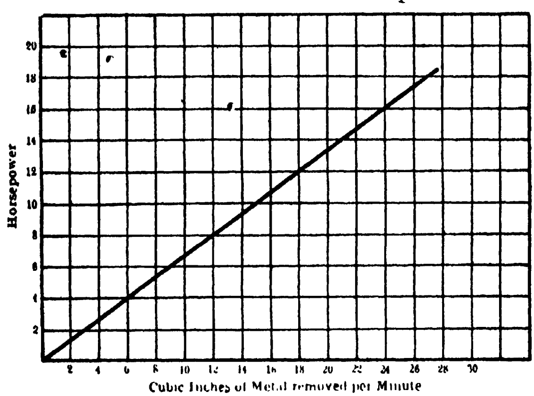


FIG. 8.—Relation between horse-power and cubic inches metal removed; mild steel, 0·40 per cent carbon.

The constant varies with the kind of metal removed.

The following values have been found by tests to exist for the horse-power required to remove 1 cub. in. of metal per min.:—

Brass and similar alloys	0·2 to 0·3
Cast iron	0·3 to 0·5
Mild steel (0·30% – 0·40% carbon) .	0·6
Wrought iron	0·6
Hard steel (0·50% carbon) . .	1·00 to 1·25
Very hard tyre steel . . .	1·50

It must be remembered that these constants represent general average conditions; considerable variation may occur where special cutting tools are used and special grades of metal are encountered.

Experiments were also made by Prof. Flather, and he found that the power required for **lathes** may be approximately expressed as follows:—

$$HP = {\cdot}021\ W \text{ for cast iron.}$$
$$HP = {\cdot}035\ W \text{ for wrought iron}$$
$$HP = {\cdot}039\ W \text{ for steel.}$$

Where W = weight of metal removed in lb. per hour.

Prof. Flather gives the following figures for power required to remove metal with a **milling** machine:—

$$HP = {\cdot}10\ W \text{ for bronze.}$$
$$HP = {\cdot}14\ W \text{ for cast iron.}$$
$$HP = {\cdot}30\ W \text{ for tool steel.}$$

Comparing these figures with those above, it will be seen that the milling cutter is much less efficient as regards power absorbed than the lathe or planer tool.

Prof. Flather's formula may also be expressed as—

$$HP = 7{\cdot}32\ SDF \text{ (for mild steel),}$$

where S = surface or cutting speed in ft. per min.
D = depth of cut in inches.
F = feed in inches per revolution.

See diagram, Fig. 9.

To Determine the Cutting Speed.—The usual procedure is as follows:—

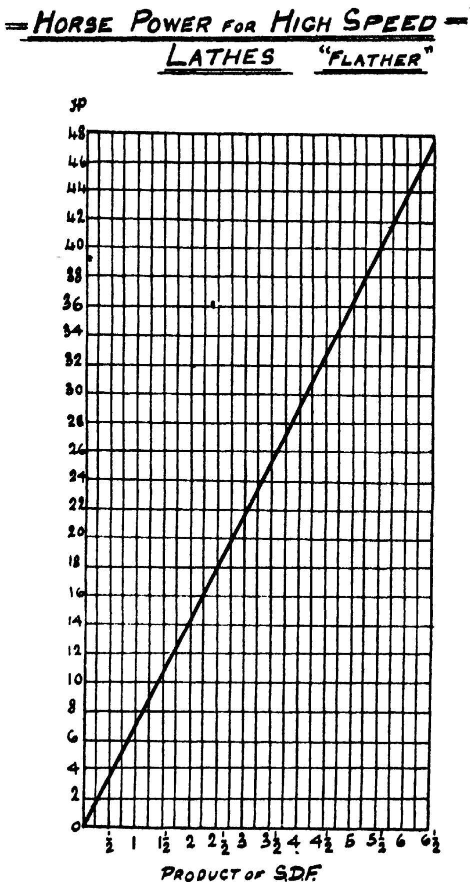

HORSE POWER FOR HIGH SPEED
LATHES
"FLATHER"
HP
48 46 44 42 40 38 36 34 32 30 28 26 24 22 20 18 16 14 12 10 8 6 4 2 0
½ 1 1½ 2 2½ 3 3½ 4 4½ 5 5½ 6 6½
PRODUCT OF S.D.F.

FIG. 9.

Cutting speed (ft. per min.)

$$= \frac{3{\cdot}1416 \times \text{dia.} \times \text{RPM}}{12}.$$

For approximate calculations which may be done quickly, the quantity 3·1416 may be considered as 3. The equation would then read

$$\tfrac{3}{12} \times \text{dia.} \times \text{RPM}$$

$$\text{and simplified, CS} = \frac{\text{dia.} \times \text{RPM}}{4}$$

$$\text{or} \quad \text{RPM} = \frac{\text{CS} \times 4}{\text{dia.}}.$$

Cutting Pressure on Tools.—The Manchester experiments conducted in 1903 on rapid-cutting tool steels showed that for steel the cutting force is simply proportional to the area of cut, and that this force has the following approximate values:—

For *soft* steel (fluid compressed) . . . 115 tons per sq. in.

For *medium* steel (fluid compressed) . . . 108 tons per sq. in.

For *hard* steel (fluid compressed) . . . 150 tons per sq. in.

For *cast* iron this stress may be specified as follows:—

For *soft* cast iron . .	51 tons per sq. in.
„ *medium* „ „ . .	84 „ „ „ „
„ *hard* „ „ . .	82 „ „ „ „

Endurance.—In the Manchester experiments trials were made to test endurance of the steels upon a lengthened run with a uniform shape of tool.

These trials showed that a speed of 90 ft. per min. may be maintained for a considerable period

with a $\frac{3}{16}$ in. cut by $\frac{1}{16}$ in. traverse upon material similar to the fluid-compressed soft steel operated upon, as only three out of seven tools failed to fulfil the expectations of the committee. They also showed that the steels cut more than twice as fast as ordinary Mushet steel, and more than four times as fast as ordinary water-hardened steel.

With regard to the endurance trials on medium cast iron, three out of thirteen tools tried ran for an hour, or longer, at 34 ft. per min. with a $\frac{3}{16}$ in. by $\frac{1}{16}$ in. cut; but no tool completed the intended run of two hours' duration.

An ordinary Mushet tool ran for an hour on this material at a speed of 19½ ft. per min., whilst ordinary water-hardened tools failed in from 4 to 9 min. at 12 ft. per min.

Power Transmitted by Belts.—When the horse-power has been ascertained it will then be necessary to find the requisite strengths of belt drive and gearing to make use of this horse-power. Various tables of the power which may be transmitted by belts have been published, but, generally speaking, if we allow a speed of 500 ft. of belt per min., 1 in. wide for 1 horse-power, we have a reliable formula which will not overstress the belt. The table on page 207 is in a very convenient form for obtaining horse-power when the speed of the belt is known. This speed is obtained by the formula for cutting speed on page 25.

Strength of Spur Gear Teeth.—The formula for strength of spur gear teeth usually accepted in machine tool practice is that of Mr. Wilfred Lewis. This is

$$W = SPFy,$$

where W = working load on teeth in lb.,
S = safe stress of material in lb.,
P = circular pitch of teeth in ins.,
F = width of face of gear in ins.,
y = factor for strength depending on the form of tooth and number of teeth in the gear. Pressure angle is assumed at 15°.

For convenience, when dealing with diametral pitches, this formula may be written—

$$W = \frac{SFy\pi}{D},$$

where D = diametral pitch,

$$\pi = 3{\cdot}1416.$$

The safe stress varies, of course, with the material, and it also varies (in this formula) with the speed at which the teeth run—the greater the speed the smaller the stress. This feature has some effect on the durability of the gears.

The factor y allows for the variation in strength of gears of the same pitch, but with different numbers of teeth. As is well known, the rack tooth of any series is the strongest tooth, and a twelve-tooth pinion the weakest.

The accompanying table is that used with the Lewis formula.

TABLE FOR y.

No. of teeth .	12	13	14	15	16	17	18	19
y . . .	·067	·070	·072	·075	·077	·080	·083	·087
No. of teeth .	20	21	23	25	27	30	34	38
y . . .	·090	·092	·094	·097	·100	·102	·104	107
No. of teeth .	43	50	60	75	100	150	300	rack
y . . .	·110	·112	·114	·116	·118	·120	·122	·124

TABLE FOR S (ALLOWABLE STRESSES).

Velocity in Ft. per Min.	100.	200.	300.	600.	900.	1200.	1800.	2400.
Cast-iron	8000	6000	4800	4000	3000	2400	2000	1700
Steel .	20,000	15,000	12,000	10,000	7500	6000	5000	4300

The subject of the strength of spur gear teeth is one on which there is much diversity of opinion. The author made out a table for handiness as given below. It is based on the results of observation of various gears at work and not on any scientific tests. Speed and form of tooth are not taken into account, these being dealt with by the Lewis formula, and those who wish to go into refinements may use both conjointly, but it must be always borne in mind that *the strength of any train of gearing is determined by the strength of the smallest pinion in the train.*

SUGGESTED SAFE PRESSURES PER INCH OF WIDTH.
CAST IRON. CUT.

Pitch.	Pressure in Lb.	Pitch. In.	Pressure in Lb
10	250	$\frac{7}{8}$	700
8	300	1	800
6	400	$1\frac{1}{8}$	900
5	500	$1\frac{1}{4}$	1000
4	600	$1\frac{1}{2}$	1200
		$1\frac{3}{4}$	1400

The tables of diametral pitches given in the Appendix will be found useful in this connection. They were originally published in "Machinery".

Let us apply the Lewis formula to a definite case.

The table of a 48 ins. boring and turning mill has a gear 100 T, $2\frac{1}{2}$ DP, $3\frac{1}{2}$ ins. wide, and is 40 ins. diameter. At the slowest table speed, the speed is 1·7 rev. per min., the teeth therefore having a periphery speed of

$$\tfrac{40}{12} \times \pi \times 1{\cdot}7 = 17{\cdot}8 \text{ ft. per min.}$$

Looking in the column of safe stress, we see that $s = 8000$ lb. for cast iron. The corresponding circular pitch to $2\frac{1}{2}$ DP is 1·257.

$$f = 3\tfrac{1}{2} \text{ ins., and } y = {\cdot}118;$$

$$\therefore \text{W} = 8000 \times 1{\cdot}257 \times 3{\cdot}5 \times {\cdot}118 = 4153 \text{ lb.}$$

The horse-power transmitted would thus equal

$$\frac{4153 \times 17{\cdot}8}{33000} = 2{\cdot}24 \text{ HP.}$$

We must, however, take into account the probable number of teeth which are taking the load—in this case at least two, and probably three—and as the Lewis formula gives the load for one tooth, we should then get $2{\cdot}24 \times 3 = 6{\cdot}72$ horse-power.

We now take another factor into consideration, and, neglecting speed, compare the strength of tooth with the probable pressure on the tools working on the largest diameter. The Manchester experiments with high-speed steel showed that the cutting force is proportionate to the area cut, and that this force for hard cast iron is 82 tons per sq. in. (see page 25). Assuming two tools to be each taking a cut of

$$\tfrac{1}{4} \times \tfrac{1}{16}, \text{ we have } \tfrac{2}{1} \times \tfrac{1}{4} \times \tfrac{1}{16} = \tfrac{1}{32} \text{ sq. in.,}$$
$$\text{and } \tfrac{82}{32} = 2{\cdot}56 \text{ tons} = 5740 \text{ lb. pressure.}$$

As the table gear is smaller than the largest diameter admitted, the pressure on the teeth will be

increased in the ratio of 48 to 40; we then get

$$5740 \times \tfrac{48}{40} = 6888 \text{ lb.}$$

We saw that the tooth had a strength of 4153 lb., so that with either two or three teeth in contact, there would be an ample margin of safety.

Selection and Care of Gears.—The width of face of a spur gear varies from two to six times the circular pitch and is governed by the type of machine on which it is to be used. If made narrow, a greater pitch is required to give equivalent strength, but it should be borne in mind that a fine pitch and a wide tooth will do more work, induce smoother running, and ensure better all-round results (especially at high speeds), than a coarse pitch and a narrow tooth. Teeth narrow in proportion to the pitch have not the same rigidity and power to resist shocks. This can easily be seen by the excessive vibration set up by narrow teeth as compared to the others.

In order to obtain the best results the gears and their pinions should have machine-cut teeth, accurately corresponding in size and form to the tooth profile of their respective diameters. Special care should also be taken to see that the gear blanks are turned up true and to the correct outside diameter, according to the recognized standard. Each pair of gears must be adjusted to the correct centres according to their diameters, so that the pitch circles of the two gears intersect in running. The importance of accurate mounting cannot possibly be overestimated. No matter how well and accurately gears are made, they cannot give complete satisfaction

unless they are correctly mounted and kept running true. Proper consideration and care given to this matter at the right time will often save very considerable subsequent trouble and possible loss. The alignment of the two shafts must be perfect, and their bearings must be stayed together most rigidly to keep them exactly the right distance apart.

Gear wheels are more often cut too deep than not deep enough, and, as far as noise is concerned, it is worse to have the driver too deep than the driven gear. Another cause of noise may be that the cutting is not central, as shown by gears being noisy when running one direction and quiet in the reversed direction. Again, the centre distance may not be right; if meshing too deep, the outer corners of the teeth of one gear may strike hard against the roots of the teeth of the other gear. Still another cause of noise may be found in the fact that the frame carrying the gear shafts is of such a form and size as to give off sound vibrations.

A pinion should never be used with less than twelve teeth, and preferably not less than fifteen. No pinion working at a high speed should have less than twenty teeth at the very least. If made with a smaller number of teeth, the form of the latter necessarily becomes distorted (through under-cutting), and their strength materially decreased; also, the tooth area in gear at one time is proportionately lessened, a greater load therefore falling on each tooth. In addition to this, the shocks—although decreased in number—are proportionately increased in strength, conducing to an increase of vibration, and to less even running. These combined circumstances render the use of

pinions with a small number of teeth most undesirable for high speed work, placing their probable working results altogether very much in doubt, and making it practically impossible to form any definite theoretical estimate of their actual capacity.

Further, it is advisable that whenever possible a pair of gears should be *prime to one another*, so that the same pair of teeth are not in mesh continually with each other.

Materials for Gears.—The selection of suitable materials for gears is usually determined by the character of the mechanism and its uses. Some machines permit the use of gears of large dimensions, while in others the sizes are often very limited. Hard, close-grained cast iron can be used in all places where there is sufficient room for the proper dimensions, or for gears that are employed only part of the time the machine is working, and are engaged and disengaged when the machine is stopped, such, for instance, as the back gears for a lathe—where they can be made of large diameter, wide face, and run at moderate speeds.

If, however, a large amount of mechanism must be placed in a small compass, such as change-speed gear boxes, with the changes made by handles while the gears are running, or if the gears are subject to heavy loads or shocks, then cast iron is unequal to the requirements, and either mild steel or alloy steel should be used — case-hardened or heat-treated.

The Lewis formula with its modifications is a safe guide in selecting the materials, and cast iron or steel gears proportioned with this formula should

preferably *not exceed* 1200 *ft. per min.* pitch-line velocity.

When not overloaded, the teeth of cast-iron gears wear to a polished and glazed surface so characteristic of this material. . When slightly overloaded, the wearing surface of the teeth becomes crushed and cut at the pitch line and then wears very rapidly; when considerably overloaded, failure occurs through breakage.

Semi-steel, as a mixture of steel and cast iron is sometimes called, but which is really a high grade of cast iron, is an improvement over ordinary cast iron, being strong enough to carry about 50 per cent more load, and giving similar results in wear and breakage.

Mild steel gears, having 0·40 to 0·50 per cent carbon, will carry from 2 to 2½ times the load for cast iron; when slightly overloaded they wear a line depression across the face of the teeth at the pitch diameter, and when continuously overloaded soon wear out. Steel gears containing about 0·20 per cent carbon can be case-hardened after being machined; when they will stand considerable use without wear and are especially valuable for pinions meshing into large gears. The question of hardened versus unhardened gears is, therefore, one of space and weight limitations. It should be noted that cast iron running with cast iron, or a steel pinion running with a cast-iron gear is a good combination, but that two mild steel gears running together, particularly at high speeds, should be avoided under any circumstances. The soft steel gears then abrade with great rapidity.

Whatever material is adopted, it is advisable to make the pinion of harder material than the gear in order to divide the wear equally between the two, and thus prolong their useful life by enabling the teeth of both gear and pinion to retain their original accuracy and shape as long as possible. If undue wear affects the pinion, this in turn reacts upon the teeth of the gear, and both suffer.

Heat-treated Alloy Steel Gears.—Unlike case-hardened gears, heat-treated, or tempered, gears are of uniform carbon content throughout, and have a uniform hardness throughout the tooth section. The steels used for heat-treated gears are of three general types—silico-manganese, chrome-vanadium, and chrome-nickel steel—the last named, in its several modifications, being by far the most used. The carbon content varies, for the different types, from 0·40 to 0·60 per cent. The heat treatment of all these types is simplicity itself, consisting merely in heating the gear slowly and uniformly to the hardening temperature, which is usually about 1500° F., and quenching in oil, after which it is drawn in an oil bath. The result is a strong, tough, dense-grained steel gear, which has been used with marked success in motor-car work, and which is fast replacing mild steel and case-hardened gears in machine tool construction.

Viewed from the standpoint of physical properties in the finished gear, the evolution in gear material from cast iron to heat-treated steel may be seen in this table:—

EVOLUTION IN GEAR MATERIALS.

Material.	Tensile Strength. Lbs. per Sq. In.	Hardness Scleroscope.	Toughness.
Cast iron	20,000	25	Negligible
Mild steel	45,000	35	2
Case-hardened steel (average test of alloy steels)	120,000	85	2·5
Heat-treated steel (average test of alloy steels)	225,000	75	5

For machine tools, heat-treated alloy-steel gears appear preferable to case-hardened gears for a number of reasons.

Physically they are stronger and tougher and should, therefore, be better able to resist sudden impacts and extraordinary loads. Heat-treated gears do not show by file and scleroscope test the same degree of hardness as case-hardened gears, but nevertheless with proper design, the dense-grained, tempered gear-tooth resists wear most satisfactorily.

Used in gear boxes, especially for sliding gears when changing speeds while running, case-hardened gears are liable to have the hard case chipped off, thereby exposing the soft core to the impact of clashing. The hard chips fall into the gearing and may find their way into bearings with resultant troubles. Heat-treated or tempered gears with a uniform hardness throughout do not chip.

The heat treatment of such gears is much simpler than that required for proper case-hardening. It is shorter, less costly, and produces a more uniform product, and the gear as compared with case-hardening, freer from warpage, a point of especial value for quick-running gears.

CHAPTER II.

APPLICATIONS OF THE CONE PULLEY.

THE plan of the fast headstock shown in Fig. 10 is a representative example of good practice in regard to small lathes, and may be taken as a type of a driving headstock generally, from which motion may be taken at either end of the driving shaft. The cone pulley A runs loose on the driving shaft (or spindle) and receives its motion by belt from an overhead counter-shaft. It is of hollow construction for lightness and is supported on the shaft at one end by a plate B, fastened to it by cone-head screws, and at the other end by a pinion D. When it is desired that the driving shaft shall run without the back-gear reduction, the locking bolt C is lifted up till it fits a notch in the plate B. This bolt is carried by the gear G, keyed to the shaft, so that the cone pulley, plate, gear, and shaft, rotate together as one piece, and at four different speeds. When required to run slower, the back gears would be brought into operation, and the speed reduced in the ratio of the back gears. The gears would be in mesh as in the illustration, with the locking bolt free from the notch in plate B, and the motion would then be transmitted through the cone pulley, pinion D, gear E, pinion F, and gear G.

The most common practice with the back shaft carrying the gears E and F, is to make this as a tube revolving on a stationary shaft, and fitted with

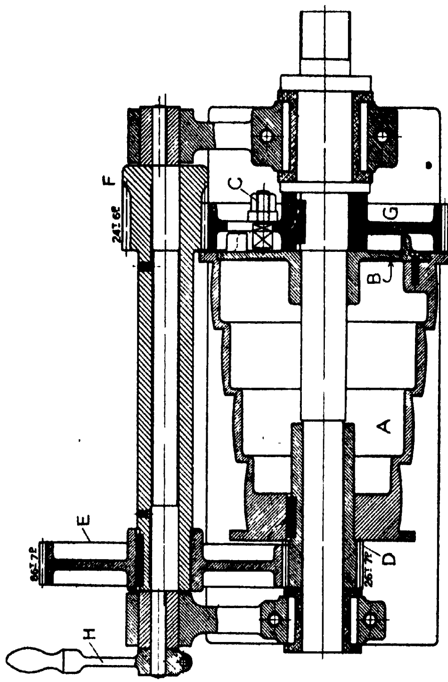


FIG. 10.—Ordinary back-geared head.

an eccentric portion at each end, either on the shaft or the bushes. By this means, and giving the handle H a half-turn, the gears on the back shaft may be withdrawn out of mesh with the other gears when not required for driving.

Ball-bearing Applications.—The lathe head

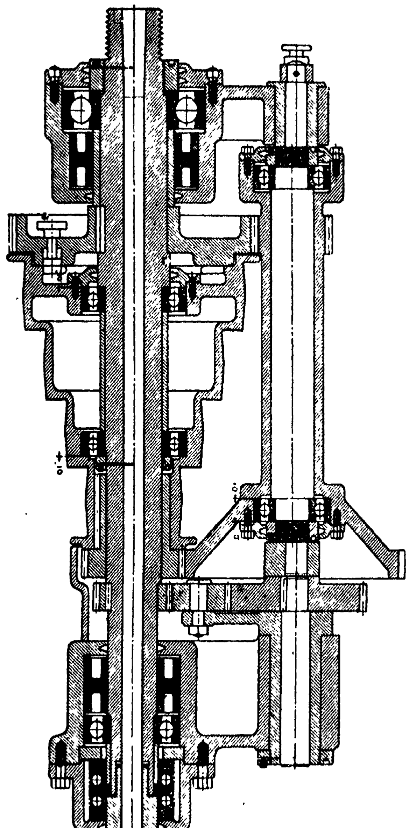


FIG. 11.—Back-geared head showing application of ball bearings.

shown in Fig. 11 is an interesting example of all-ball-

bearing design by the Hess-Bright Manufacturing Co. The spindle is located endwise by a double-thrust washer at the left-hand end. The cone pulley and back gears are located by journal bearings with ·01 inch end play.

Double Back Gears.—As pointed out in the pre-

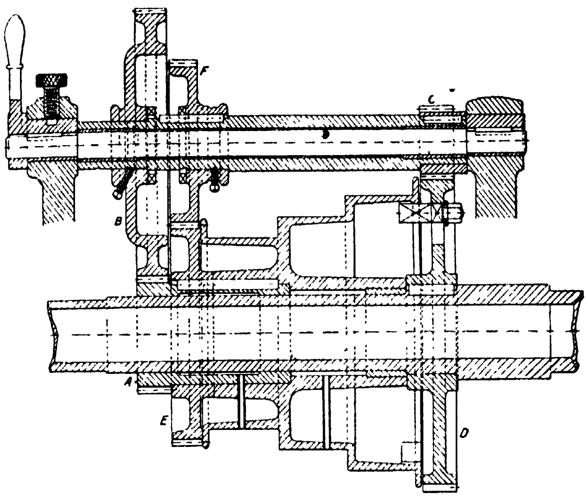


FIG. 12.—Head with three-step cone and double gear.

ceding chapter, it is an advantage to have fine steps in graduations of speeds, and a headstock fitted with an extra pair of gears, or double back gear, as it is termed, enables this to be done, and also results in the use of a cone with larger diameters of the steps, as was shown in Fig. 5, and the variation in power reduced to a minimum. In Fig. 12 the construction of such a head is shown, as made by Messrs. Tangyes

Ltd., Birmingham. This has a three-step cone and with the use of a two-speed counter-shaft a total of eighteen spindle speeds are obtained. The three steps taking a 4-in. belt, are 11 ins., 13½ ins., and 16 ins. diam. The arrangement of the double gear is very ingenious and interesting. The slow speed is from the pinion A to gear B on the back-gear tube, and from pinion C to large gear D on the spindle. This all as in the previous case, but now the additional pair of gears have to be taken care of, for these should not be in mesh unless actually transmitting motion, so the gear B is not keyed to the eccentric tube. The centre is made in cup form to allow the gear F to pass inside, and out of mesh with gear E. The gear F is keyed to the tube and is provided with clutch teeth, which engage with corresponding clutch teeth on B, and the motion from B is thus transmitted to the tube. Room is left on the tube so that B may be moved out of mesh with A, when E and F are in mesh. The set screws are for holding the gears in whatever position they are placed.

In comparing single and double back-gear heads for relative power transmission, equal diameters of the largest step of the cone and equal ratios of speed range should always be taken to allow of a fair comparison. The horse-power put into a machine is governed by the number of square inches of belt passing over the pulley per minute. This is at its maximum in a cone drive at the point where it is usually least needed, that is, at the highest spindle speed, on work of small diameter; and at its minimum on the lowest spindle speed, on work of large diameter. Therefore the lowest spindle speed is the

weak point in the speed range, and comparison should be made relative to this. Take, for instance, examples in Figs. 13 and 14. Here, a speed ratio of 20 to 1 is assumed, with twelve speeds. The

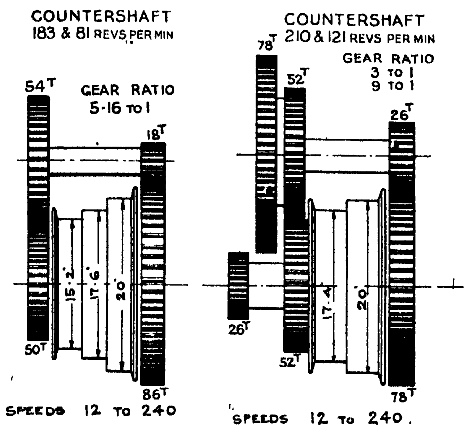


FIG. 13. FIG. 14.

Comparison of single and double back gear.

counter-shaft speeds specified give 12 to 240 revs. per min. Twelve speeds are used because this number can be obtained under the same conditions, that is, using two speeds to the counter-shaft in each case. On the lowest spindle speeds, the speeds of the belts are $\frac{15{\cdot}2 \times \pi \times 81}{12} = 323$ ft. per min., and $\frac{17{\cdot}4 \times \pi \times 121}{12} = 552$ ft. per min. respectively, so that the double back gear shows an advantage of 71 per cent over the single gear. As

the space occupied by each arrangement does not vary greatly, the increase in power warrants its adoption.

Clutches.—On many classes of work it is essential that means are provided for the quick changing of speeds, especially from single to back gear and vice versa, in order to suit various types of tools. This is usually accomplished by the introduction of friction clutches to engage the various rotating members, these being operated by handles in a position convenient for the operator, and the ordinary locking bolt between the cone and adjoining gear dispensed with.

Where a *cone clutch* is used, it should be remembered as a golden rule that "It must be locked up against a shoulder on the piece it is to rotate"; otherwise, if it is against an immovable piece, it will act as a brake and stop the motion entirely. It should also have self-contained on the same rotating piece something positive to back it up, and prevent it slipping away from engagement. A cone clutch should be positive in its gripping power and at the same time easy to release without sticking when the force pressing the two parts together is released. This greatly depends on the angle. With a small angle the cones will seize and not release, and with a large angle too much pressure is required to keep the parts in contact. An included angle of 25°, or 12½° each side, has been found to be the most satisfactory in practice to fulfil the above conditions. It is also found an advantage to drill holes in the body of the clutch to allow for the escape of air, and so avoid a cushion being formed which would tend to prevent perfect contact of the gripping surfaces.

Likewise, three or four spiral grooves chipped across the external surface of the cone are also found to give very good results. Oil coming on to the conical surfaces is not detrimental, but rather an advantage, forming a lubricant when running freely. The action of the cones coming together is to force all lubricant out.

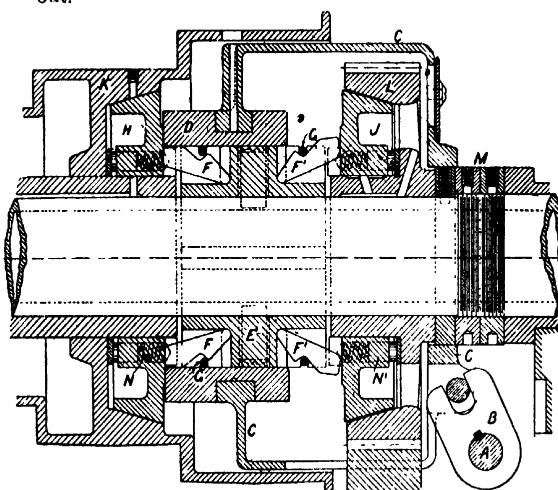


FIG. 15.—Example of cone clutch.

An ingenious arrangement for operating cone clutches is shown in Fig. 15. This is an example from the back gear of a lathe, and the clutch operating device is unique in several ways. A hand lever, not shown, is keyed to the shaft A and causes a movement to be given to the lever B, and from this to the cage C. This cage C serves two purposes. It

forms a guard over the gears, and moves the sliding collar D. On the spindle, and secured to it by two screws to prevent endwise movement, is a piece E, carrying in slots at each side six little steel wedges or toggles, F, F', which are prevented from flying out by a spring wire G. These toggles also fit in corresponding grooves cut in the backs of the cones H and J, and thus act as keys to connect them to the piece E, and thence to the spindle. As shown in the illustration the collar D has been moved to the left hand, and riding up the inclines on the toggles F, has caused these to be depressed, and practically lengthened. The depressing of these levers causes great pressure to act behind the cone clutch H, and forces it into contact with the internal cone formed in the cone pulley. The cone pulley then drives the spindle direct. So long as the toggles are covered by the sliding collar, they are held down to their work, so fulfilling the requirements laid down as a golden rule. When it is desired to drive through the "back gear" the sliding collar D is moved to the right, uncovering the toggles F, which spring outwards and release the clutch H. Then the opposite toggles F' are depressed, causing the clutch J to come into operation and drive the spindle at a reduced speed through the back gearing. It will be at once noticed that with such an arrangement, the piece E having to receive pressure on both sides must be rigidly fixed to the spindle, and to obtain the maximum effort exerted by the toggles that the internal cones K and L must be provided with a fine adjustment. In this example lock nuts are provided for each clutch, one set being shown at M. Springs

are placed in the cones, as at N, N′, to force them apart when released.

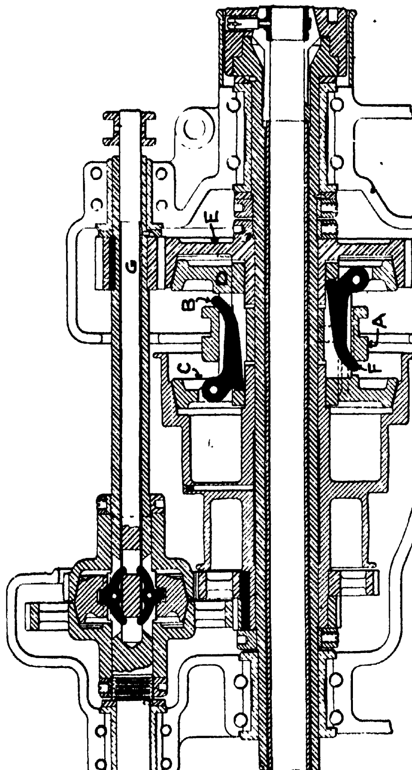


FIG. 16.—Head with cone clutches.

Another interesting example of conical clutches is shown in Fig. 16, being one of a three-speed cone and

double back gear. The combination gives nine spindle speeds, with three speeds available by means of levers and without shifting a belt. The method of moving the conical pieces should be noted. Taking the spindle, the sliding collar A when moved to the right would depress the end of the lever B pivoted to the cone C ; cone C is attached rigidly to cone D, and therefore as the short end of the lever B presses against a steel piece fixed to the spindle, the cones C and D are constrained to move to the right and thereby engage the gear E. When the sliding collar A is moved to the left, it acts upon the lever F to first withdraw the clutch from the gear, and then engage it with the cone pulley. The back-gear clutches are operated by the end movement of the rod G, which acts upon the curved ends of little levers pivoted to the back-gear shaft, thereby causing the centre sliding cone to be forced into either of the two gears in accordance with the direction in which the rod is moved.

Expanding Ring Clutches are frequently used in preference to conical clutches, and there are many and various designs of same. Generally, they are more reliable than conical clutches. As their name implies, they consist primarily of a split ring with means for expanding same. One type very commonly used in conjunction with cone pulleys is that in which a toggle joint is used to cause the expansion of the ring. The details given in Fig. 17 show very clearly the arrangement and working of such a joint. A is the expanding ring and B a gear to which it is to be connected when expanded. It will be seen that the ring is divided and provided with enlarged

ends. The remaining portion of the ring is complete, and not cut away again.

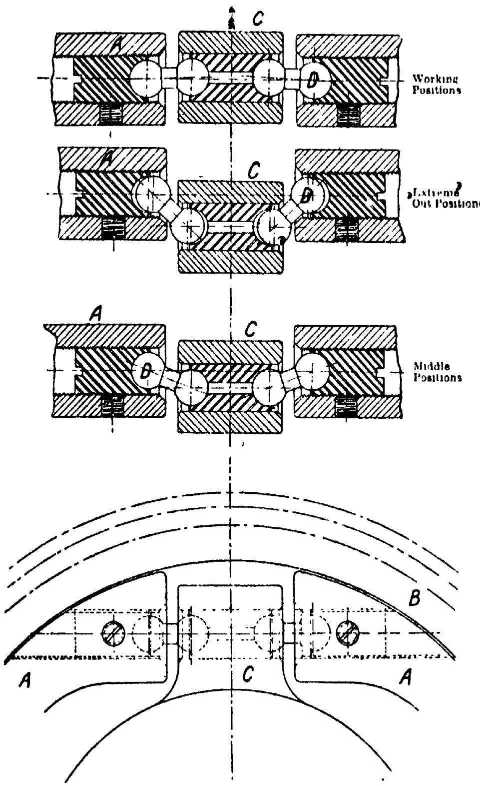


FIG. 17. -Toggle joint clutch.

Between the ends of the ring there is placed a projecting piece C, which is part of a sliding collar

moved by the operating lever, and concentric with the ring. In C there is placed a hardened steel bush, and in A there are hardened steel bushes screwed on the outside, so that they may be adjusted. A grub screw locks each in position. The connection between the piece C and the ring is made by two short rods D, having ball-ends so that they may work freely in the bushes. Of the three positions of the toggles shown, the lower one shows the position when the clutch is disengaged. The sliding collar is intended to work two clutches, as in the case of a cone and back gear, and the middle sketch shows the toggles when the sliding collar has been moved to put the opposite clutch into gear, and the toggles shown are therefore in extreme out position. The topmost sketch shows the toggles in correct working position.

To expand the ring, the movement of the sliding collar would be in the direction of the arrow. It should be particularly noticed that the toggles have just *passed over the point where they would be in one straight line*, or the line of maximum effort. By so adjusting them, they will expand the ring and keep in position of themselves without any other means of holding; but if they do not quite come up to the straight-line position, then they will try to work backwards, and become disengaged. These clutches are very powerful when properly fitted.

In Fig. 18 is shown the usual arrangement of these clutches, and also the proportions of a series arranged and used by the author.

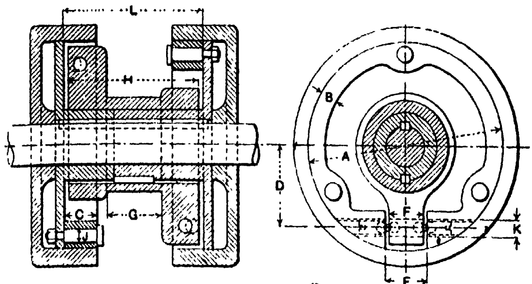


FIG. 18.

PROPORTIONS OF EXPANDING RING CLUTCHES.

A	B	C	D	E	F	G	H	J	K	L	Movement of Slider.
in.	in	in.	in.	in.	in.	in.	in.	in	in.	in.	in.
7	$\frac{9}{16}$	$1\frac{3}{16}$	$2\frac{3}{4}$	$1\frac{1}{2}$	$1\frac{1}{4}$	2	$4\frac{9}{16}$	$\frac{1}{2}$	$\frac{5}{8}$	5	$\frac{3}{16}$
$8\frac{1}{4}$	$\frac{13}{16}$	$1\frac{1}{4}$	$3\frac{3}{8}$	$1\frac{5}{8}$	$1\frac{3}{8}$	2	$4\frac{3}{4}$	$\frac{1}{2}$	$\frac{5}{8}$	$5\frac{1}{8}$	$\frac{3}{16}$
$9\frac{1}{2}$	$\frac{7}{8}$	$1\frac{3}{8}$	$3\frac{7}{8}$	$1\frac{7}{8}$	$1\frac{5}{8}$	$2\frac{1}{4}$	$5\frac{1}{4}$	$\frac{9}{16}$	$\frac{3}{4}$	$5\frac{3}{4}$	$\frac{7}{32}$
$10\frac{3}{4}$	$\frac{7}{8}$	$1\frac{7}{16}$	$4\frac{1}{4}$	$2\frac{1}{8}$	$1\frac{3}{4}$	$2\frac{3}{8}$	$5\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$6\frac{1}{8}$	$\frac{1}{4}$
13	$\frac{7}{8}$	$1\frac{1}{2}$	$5\frac{1}{4}$	$2\frac{3}{8}$	$1\frac{7}{8}$	$3\frac{1}{8}$	$6\frac{3}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	7	$\frac{1}{4}$
$14\frac{1}{2}$	1	$1\frac{1}{2}$	$6\frac{1}{16}$	$2\frac{3}{8}$	$1\frac{7}{8}$	$3\frac{1}{8}$	$6\frac{1}{2}$	$\frac{11}{16}$	$\frac{3}{4}$	7	$\frac{1}{4}$
$16\frac{3}{4}$	$1\frac{1}{4}$	$1\frac{5}{8}$	$7\frac{1}{8}$	$2\frac{3}{8}$	$1\frac{7}{8}$	$3\frac{1}{4}$	$6\frac{3}{4}$	$\frac{11}{16}$	$\frac{15}{16}$	$7\frac{7}{16}$	$\frac{5}{16}$

Treble Gears.—When it becomes necessary owing to the class of work to be operated upon that a slow speed of revolution is required, then additional pairs of gears are added to the train of gearing from the first driving shaft to the final. Such gearing is known as treble and quadruple gearing, and in exceptional cases quintiple gearing is used. A treble gear is one in which the velocity ratio is increased in three steps, or by the interposition of three pairs of gears between the cone and the

spindle. The most common form of treble gear arrangement is shown in Fig. 19. There is the usual four-step cone pulley and back gear, giving eight changes of speed to the spindle, and then an additional set of four speeds through the treble gear. As shown the gears are set for treble gear.

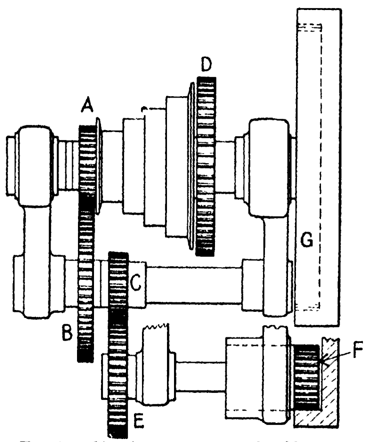


FIG. 19.—Simple arrangement of treble gear.

When running without gear, the gears B and C are disengaged by an eccentric, pinion F is withdrawn into its bearing out of mesh with the faceplate gear G, and the cone pulley locked to the spindle.

For the double gear, A drives B, and C is moved along the eccentric tube to mesh with and drive gear D; and for the treble gear, A drives B, C drives E, and F drives G, the cone running free.

It will be noticed that gear G is an internal gear, necessary to bring the direction of rotation in the right direction as shown in Fig. 20.

Proportions of a treble gear taken from an actual example are as follows:—

Cone diameters, 23 – $19\frac{1}{2}$ – 16 – $12\frac{1}{2}$ ins., $5\frac{1}{2}$ ins. wide; double-gear ratio, 11·45 to 1; treble-gear

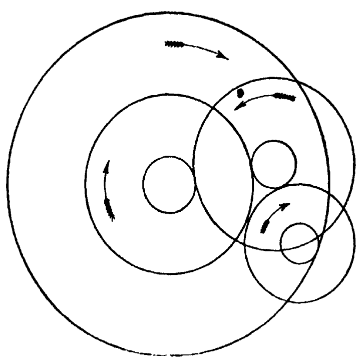


FIG. 20.—Diagram of treble gear.

ratio, 59·3 to 1. There are twelve spindle speeds, with a single counter-shaft speed of 200 R.P.M.

No gear.	Double gear.	Treble gear.
368, 243, 164, 108	32, 21, 14·3, 9	6·2, 4·1, 2·6, 1·8.

It will be noticed that the steps are not strictly in geometrical progression, as some designers prefer to have the greater number of speeds on the slow ranges, where changes in diameters of work very quickly alter the surface speed, and the finer steps on the slow ranges have thus an advantage.

In Fig. 21 is shown a similar arrangement of treble gear with the addition of another pair of back gears. This has the advantage previously mentioned of a better cone pulley, and also a finer speed range, there being two double gears and two treble gears, five sets in all. The gear runs are (1) HJCD; (2) ABCD; (3) HJCEFG; (4) ABCEFG.

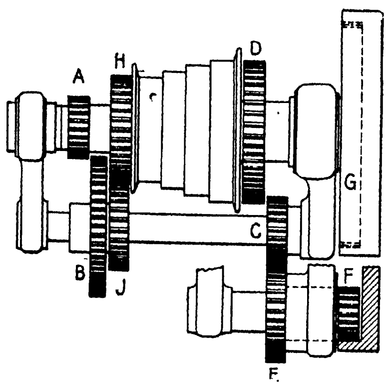


FIG. 21.—Treble gear with double back gear.

Where very slow speeds are required for heavy work it is better to remove the cone from the spindle and place it on a separate shaft, and drive the spindle through at least one gear ratio. The effect of this is that a much larger cone pulley can be used with a correspondingly higher belt speed, further increased by the gear ratio. Consequently a much lighter belt can be used; or for any given width of belt, the head becomes much more powerful. Messrs. J. Lang & Sons use this method

extensively on their surfacing and boring lathes, a typical example being shown in Fig. 22. The large cone is geared to the main spindle in the ratio of

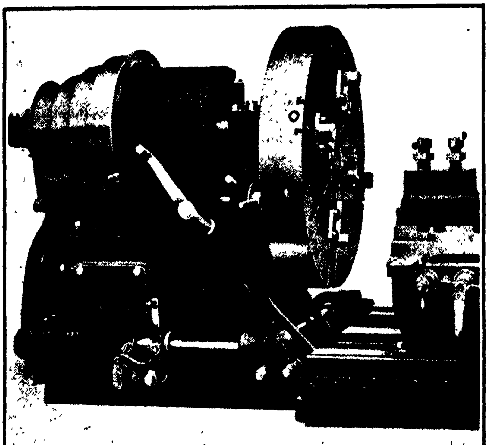


FIG. 22.—Headstock of Lang's 6 ft. surfacing lathe.

6 to 1, and further ratios of 23 to 1 and 90 to 1 may be obtained. The gears are enclosed by a guard, and changes of gear are made by the handle in front. The speed of the counter-shaft is 300 R.P.M., and there are twelve spindle speeds ranging from 2 to 83·3 R.P.M.

In Fig. 23 is shown in diagram another arrangement of gear with the cone on a side shaft. There is always one ratio of gear in use, viz. between A and B, and the gears are therefore one single gear and two treble gears.

The cone and pinions C and G run together as one piece freely on the shaft, whilst gear F and pinion A are keyed to it. The gears H, D, and E are keyed upon the second motion shaft, but free to slide along same to engage their respective gears. When running single gear, the gear F is locked to the cone, and the drive is through A and B—H, D, and E being disengaged as shown.

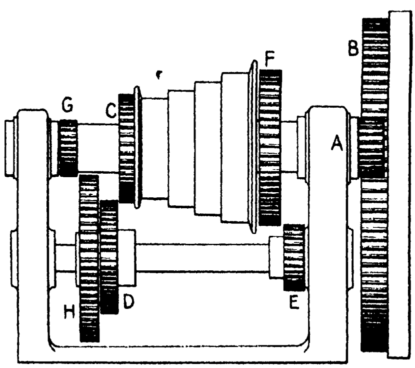


FIG. 23.—Treble gear with cone on side shaft.

In treble gear, the gear F is unlocked from the cone, E is brought into mesh, and either C and D or G and H. We then have—first treble gear, C, D, E, F, A, B; second treble gear, G, H, E, F, A, B.

In Fig. 24, representing a very large headstock, the treble gear is further extended, and a very powerful drive secured. With this arrangement there is no necessity to lock the cone as in the previous methods shown, and all changes of gear ratio are

made by sliding the respective gears in and out of mesh. There is one single gear, two double, and two treble gears. It will be seen that a third motion shaft has been provided, and three face-plate pinions fitted; by this means the wear is well subdivided. The cone and its gears are all keyed to the shaft, and the respective face-plate pinions are arranged to

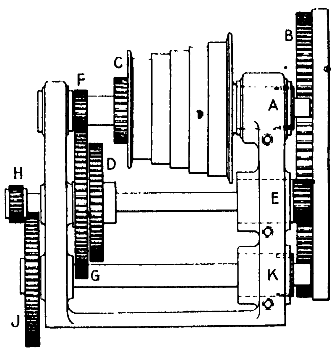


FIG. 24.—Treble gear with three face-plate pinions.

withdraw into their bearings on the head. The gear runs are: single gear, A, B; first double, C, D, E, B; second double, F, G, E, B; first treble, C, D, H, J, K, B; and second treble, F, G, H, J, K, B. It will be noticed with this arrangement that when in double gear the direction of rotation of the face-plate is reversed. It thus becomes necessary to provide the counter-shaft with a reversing motion to bring this back to the right direction.

Fig. 25 shows a headstock of a type usually used on wheel lathes. These lathes are usually double headed and the two face-plates driven from

one shaft inside the bed. Owing to the class of work they are called on to do there is a very big

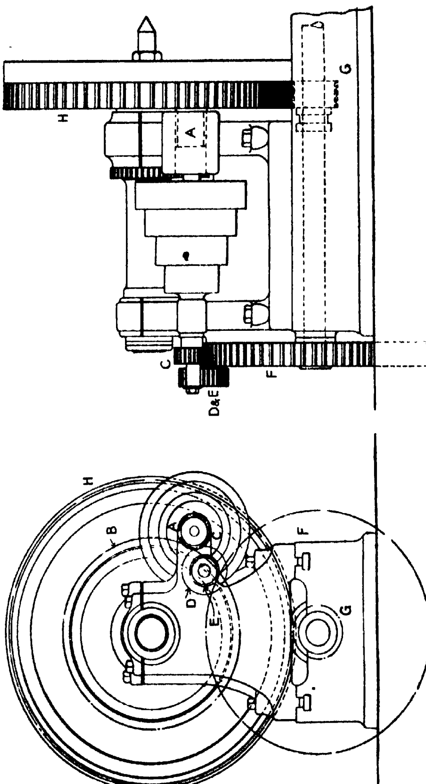


FIG. 25.—Headstock for wheel lathe.

gap in the speed range—high speeds are necessary for boring and bossing and very slow speeds for

turning the rims of the wheels. The face-plate has two external gears to allow for this, and thus only one single and one treble gear ratio are provided. The gear runs are, first through A and B for single gear, and through C, D, E, F, G, and H for the treble gear. Gears D and E run together on a stud.

Between the examples given in Figs. 19 to 25, there are many various combinations and rearrangements of the gears, due in most cases to the ratio of gearing desired, but the examples given may be taken as representing the standard designs.

Lang's Variable Speed Head.—An arrangement by Messrs. John & Lang Sons, by which a continuously variable speed can be obtained, is shown in Fig. 26.

Its object is to maintain a correct cutting speed for all diameters of work, the speed rising regularly, and not by steps as in all other designs. It will be seen that the main feature in this design is the use of two expanding pulleys A and B, which are connected by a strong belt. This belt is made by screwing a number of battens of wood to Balata belting. The ends of the battens are faced with leather and made a V-shape to suit the angle of the sides of the pulleys.

N is a loose pulley, and M the driving pulley is keyed to a shaft carrying the two expanding conical discs forming the pulley A. The two parts of this pulley are moved to or from each other by the cam C, actuated by a worm and wheel. The tension of the belt tends to force the parts of A from each other, and ball-bearings are provided at each side of C to take the thrust. A similar expanding pulley and

cam B and D are mounted on the front shaft O. One worm shaft, rotated by the hand-wheel E, operates both pulleys, so that as one opens the other closes. Thus A may, by the use of the belt touching at its edges, be made to drive B at varying

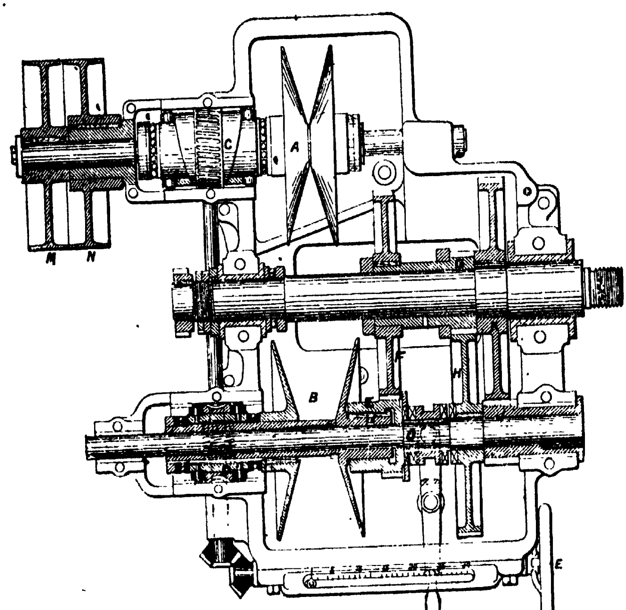


FIG. 26.—Lang's variable speed head.

speeds. To B is keyed the pinion E, which with gears F, G, and H forms an ordinary back gear, gears F and G running loose on the spindle. On shaft O is a clutch, and according to the position of this the

spindle may be driven through gears J and K, either direct or through the double gear. The maximum ratio of each expanding pulley is 3 to 1, giving a total range of speed equal to 9 to 1. The single-gear ratio is 3·9 to 1, and the treble-gear ratio 26·9 to 1.

When fitted to a surfacing and boring lathe, the hand-wheel shaft is connected to the cross-feed motion, so that when surfacing such work as cylinder covers, face-plates, etc., the revolutions of the spindle automatically increase as the diameter being turned becomes smaller, thus maintaining an approximately correct cutting speed.

CHAPTER IV.

THE ALL-GEAR DRIVE.

Advantages of the All-Gear Drive.—The cone drive was made originally to give a variety of speeds and because of this variety a long length of cone was required. Necessarily the width of the belt was cut down, because to make a cone pulley with a belt of sufficient width to give great power would not only make it difficult to shift from one speed to another, but would increase the length of the driving head-stock to undue proportions. This always limited the amount of power that could be obtained through the belt. The design, however, served its purpose up to the time of the advent of high-speed steels, motor driving, and the change in the architecture of buildings for machine shops. The modern machine shop has a very much higher ceiling than was formerly the case, and this increased the difficulty of manipulating the belt from one step to another of the cone pulley. The cutting of metal at high rates of speed called for a different design, increasing the power anywhere from two to five times, hence the use of a gear drive and single driving pulley.

This does not by any means eliminate the cone pulley, because of the fact that there is a great deal

of work where the cone pulley could be used to most advantage.

The all-gear drive, however, has not only the advantage of great increase in power, more positive action, and greater facility of speed change, but also, if installed as belt-driven machine, it reduces to a minimum the alterations necessary if at any time the machine is to be converted to motor driving. In addition to the quickness with which the changes can be made is the obvious advantage that the belt always runs at full speed, and is not slowed down just at the time when full power is needed, i.e. when large diameter work is to be done, a defect that is inherent in cone pulleys. Consequently, at whatever speed the machine is running the full power is available at the cutting surface.

One of the chief points against the cone head as compared with the all-gear head is in the fact that the power of the belt from the counter-shaft is not uniformly transmitted. The power transmitted by the belt depends upon its pull and velocity, and the pull transmitted from the driving pulley on the counter-shaft depends upon the area in contact and the adhesion per unit of area. When the belt is on the smallest step of the cone, the velocity is least, and the pull, instead of being correspondingly greater, as ought to be the case, is reduced. There is, therefore, a loss of power, which is in many cases an objection of importance, as it should be possible to get the fullest duty out of the machine when working on all diameters. With the all-gear drive, as only one diameter of pulley is requisite, it is practicable to use a much wider belt than is possible with the

cone pulley, and the pulley being made a diameter equal to or even greater than the largest diameter of the cone pulley, the maximum belt effort is always being given out. This naturally much increases the average output of the machine. As belt shifting is done away with the life of the belt is prolonged, and the drivē is more widely adaptable than when a cone pulley is used. In certain designs of gear drives the cost of erection and maintenance of a counter-shaft and additional belts is saved, with the additions of less obstructed light and a better atmosphere. The position of the machine with regard to the main drive is not limited, as the belt in the majority of cases may run at any angle either above or below, or a quarter-twist belt may be used.

The only drawback to the general adoption of any geared head is its first cost, and this determines in many cases whether it or a cone drive shall be used; it is a matter requiring careful judgment to determine whether the results from increased output, general convenience, and the other advantages named will justify the added expense.

Ease of control unquestionably results in the rapid and economical production of work. Where the work varies considerably in diameter, frequent changes of speed will be required, and where the most efficient cutting speed can be obtained by simply moving a conveniently located handle, the work will be turned out at a maximum speed because the operator will make the necessary change of speed. If frequent shifting of belts is required, a great deal of the work will be done at less than the maximum speed owing to the extra exertion involved. With

the all-gear head, any one of the spindle speeds may be *selected* instantly—hence the name sometimes given, "Selective Head".

Power in abundance—excess of power—is an essential part of a producing machine, and in this respect the all-gear drive is ideal. The power transmission means is in the form of a good belt—a wide belt—and there should then be plenty of strength of materials between the belt and the cut to take all that the belt will pull, and the belt should pull more than the cutting tool will stand. The speeds of the belt should also be considered. Two thousand feet per minute is a good speed at which to run a belt. Belts are efficient up to a certain point, and when run faster than the critical speed they cease to be efficient and wider belts should be used. Excess of belt power should always be given. A factor of safety which gives belts longest life gives better results. A belt need not be so tight if large enough, thereby relieving bearings from undue pressure, wear, and annoyance of renewals while the machine is standing idle.

The speed of gears also should be considered. Gears running above 800 ft. per min. are liable to be noisy, whereas gears running under 800 ft. peripheral speed a min. will be quiet.

An Ideal Geared Drive should have the following features:—

The minimum number of parts consistent with efficiency, to be so arranged that the least possible number of gears should be in operation at one time; any one of the entire series of changes should be possible without any more shock or jar than with a

belt or friction drive; it should be durable, not requiring more attention than a cone drive; nor should it be more expensive as to maintenance: the design should be such that it can be easily understood and operated without requiring special skill or knowledge on the part of the operator.

Position of Driving Pulley.—In the majority of cases the driving pulley of an all-gear head is placed on the left-hand side of the head and on a shaft en-

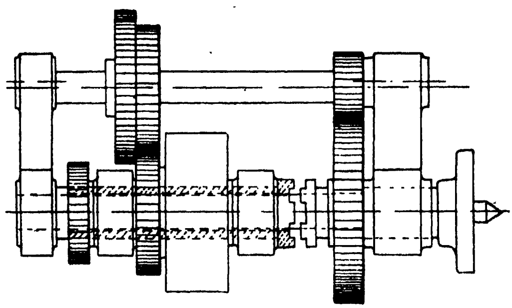


FIG. 27.—Lodge & Shipley head.

tirely away from the spindle—when used on a lathe—or final-driven shaft. It is thus in a more convenient location than when mounted directly on the spindle. In some instances, however, as in Fig. 27, it is placed with its axis coincident with the spindle, but arranged so that no belt pull comes on to the spindle or its bearings. It will be noticed that there are four distinct bearings, but of these only the two outer ones carry the spindle. The two inner ones support a sleeve which carries the driving-pulley. While the spindle passes through this sleeve it does

not touch it, so that the belt pull on the sleeve is absolutely unfelt by the spindle. On the front end of the sleeve is a clutch, and with this may be engaged a clutch which is keyed to the spindle gear, and then the pulley will drive the spindle direct. At the other side of the pulley and keyed to the sleeve are two gears of different diameters, and with these either of a pair of sliding gears on an eccentric back shaft may be engaged. There are thus three speeds in the head—one single and two double gear.

When placed on a side shaft the driving is usually arranged with a stopping motion, either in the form of fast and loose pulleys and a movement of the belt, or some form of friction clutch. Fig. 28 shows such a clutch and driving pulley. The clutch is of the disc type, in which the power is transmitted through the pressure exerted between two or more faces. A special feature in the design of clutch shown which is used on the Cincinnati milling machines, is that advantage has been taken of the cover enclosing the mechanism to make this into an additional clutch of only very light driving power, but sufficiently powerful to rotate the gearing, so that this may be always kept in motion, and changes of speed easily made. The pulley runs freely on a sleeve 1 projecting from the frame of the machine, through which passes the hollow shaft 2, having a bearing therein. The pulley carries a ring 3 attached to its web. This ring forms a clamping surface for the clutch when the pulley is clutched to the driving shaft 2. Keyed to the shaft is a frame 4, provided with two lugs, in which are pivoted the bell-crank levers 5. The collar 6 is threaded upon the periphery

of the frame 4. It is split across the centre to form a clamp, and a bolt passing through locks it in position. The clutch is operated from the controlling mechanism by a rod 7 passing through the

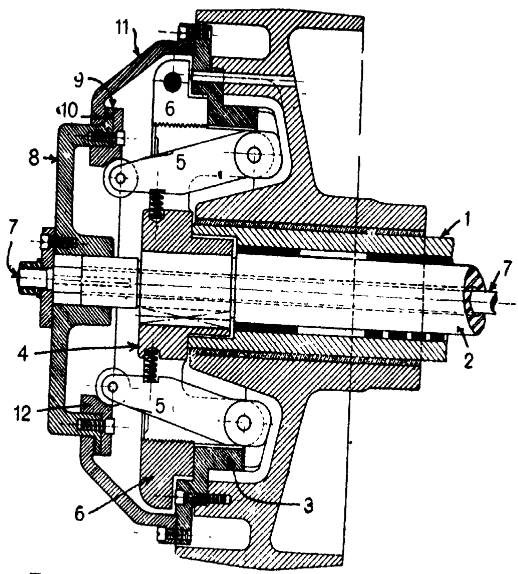


FIG. 28.—Clutch pulley used on Cincinnati milling machine.

centre of the driving shaft and connected at the outer end to the actuating disc 8, which is slidably mounted on the end of the shaft. This disc has a flange 9 upon its periphery, and against this there may be placed in contact with it a flange 10, which is part of the cover 11. This rests upon and is fastened to the

periphery of the friction ring 3. The disc 8 is provided with steel pieces 12, having coned surfaces on which rollers in the bell-crank levers 5 may travel.

Thus it will be seen that when disc 8 is moved to the right by means of the actuating rod 7, the long arms of the levers are moved inwards, bringing their short arms against the friction ring 3, compressing this against the face of the clamp nut 6, and thereby transmitting power from the pulley to the shaft. This constitutes the main drive. The auxiliary drive is imparted from the pulley to the shaft by moving the actuating rod to the left, bringing the flange 9 into frictional engagement with the flange 10. The actuating mechanism is so arranged that this auxiliary drive may be operated by the foot, and the drive depends on the pressure exerted. This leaves the operator with both hands free to move the various handles to effect the combinations of gearing.

Types of All-Gear Drives.—All-gear heads may be divided into four general types, viz :—

(1) Sliding key.
(2) Clutched gear.
(3) Sliding gear.
(4) Combination.

The Sliding-key Type has certain advantages; it is economical in space, and the speed may be changed quickly while the machine is in motion. But it is liable to get out of order, and all the gears in the train rotate, some of them idly, it is true, but it is the idle wheel, not the loaded one, that makes the most noise, and while this type may be used very appropriately for a feed motion, it is not suitable for

a main drive, and should not be used for such a purpose.

Of the *clutched gear type* we find many examples in use, with clutches of both positive and friction types. Clutches are generally in use where the number of speed changes is relatively small, and are a necessity in turret lathes and on machines where it is necessary to instantly obtain fast and slow rates of spindle speed. The clutch type has the advantage that speeds can be changed instantly without stopping the machine. For light work and high speeds friction clutches give the best results, but for heavy work with moderate speeds toothed clutches are generally used. For reliability, simplicity, and number of changes obtainable, however, there is nothing to equal the sliding-gear type—either with plain sliding gears or with tumbler-cone gearing. The combination type is, of course, endless in its variety, embracing in many instances parts of all three principal types, clutched gear, sliding gear, tumbler gear.

Clutched Gear Type.—A noteworthy practice in modern machine tool design is that of designing mechanisms as far as possible as separate units, complete in themselves, and independent of the main body of the machine. These units can then be made in quantities and put into stock, and fitted to any machine of the right type and size. A driving gear box designed as such a unit is shown in Figs. 29 and 30. It is primarily intended for belt drive, but by simply removing the lid, and replacing it with one having projecting arms, it may be easily converted to motor drive, and occupy no more floor

space than when belt driven. In making such a

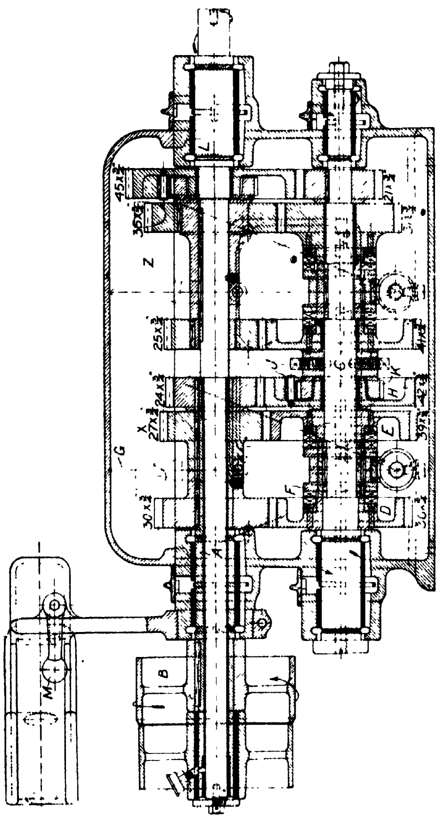


FIG. 29.—Nine-speed change gear box with clutches and silent pawl.

change the belt pulley is replaced by a chain wheel, and the drive from the motor is by silent chain.

Ordinarily a constant-speed motor is required, but if a variable-speed motor is used having a variation of only 1¼ to 1, then a practically continuous range of speeds from maximum to minimum may be obtained. In the gear box itself there are nine changes of speed, and these, combined with a double gear in

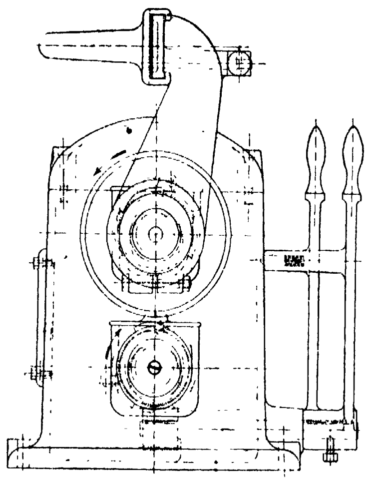


FIG. 30.

the machine, give a total of eighteen changes of speed to the spindle arranged in geometrical progression. The following table gives the ratios of gearing, and shows the spindle speeds as arranged for a high-speed radial drill, the driving pulley making 625 revs. per min. The ideal geometrical progression and actual speeds obtained by the gears are given for comparison.

Without Double Gear.

Ideal 18·6 % drop	680	554	451	367	299	243	198	161	131
Actual . .	680	568	467	370	307	254	194	161	133

With Double Gear.

Ideal 18·6 % drop	105	84·5	68·8	56·0	45·6	37·1	30·2	24·6	20·0
Actual . .	104	86·5	71·5	56·1	46·5	38·6	29·4	24·4	20·3

Ratio between highest and lowest spindle speeds . . . . = 33·5 to 1.
Double gear ratio in machine . = 6·58 to 1.
Ratio between highest and lowest speeds in gear box

$$= \frac{680}{133} = 5 \cdot 1 \text{ to } 1.$$

Or taking the number of teeth in the gears,

$$\frac{42}{24} \times \frac{45}{21} \times \frac{30}{36} \times \frac{41}{25} = 5 \cdot 1 \text{ to } 1.$$

$$\frac{\text{Maximum}}{\text{gear ratio}} = \frac{\text{driven } 42 \times 45}{\text{drivers } 24 \times 21} = 3 \cdot 75 \text{ to } 1.$$

$$\frac{\text{Minimum}}{\text{gear ratio}} = \frac{\text{driven } 36 \times 25}{\text{drivers } 30 \times 41} = 0 \cdot 73 \text{ to } 1.$$

$$\frac{\text{Maximum}}{\text{Minimum}}\ \frac{3 \cdot 75}{0 \cdot 73} = 5 \cdot 1 \text{ to } 1.$$

With this gear box the nine changes of speed are obtained with the use of only two levers, and the system of silent pawls and ratchet wheels is adopted. From examination of Fig. 29 it will be seen that

there are two sets of gears with six gears in each set. The first set X is driven by a sleeve A, to which is keyed the driving-pulley B. These mesh with three gears on the lower shaft C, on which they run freely. The gears D and E have steel clutches of the saw-tooth type secured to them, and either may be connected to the shaft C by means of the sliding clutch F, controlled by the lever G. The gear H, the slowest running of the set, carries a ratchet pawl J, and when the sliding clutch F is in the middle position, then as the motion of shaft C would tend to stop, the pawl would come into contact with the ratchet wheel K, and thus would drive the shaft. If either clutch is engaged, however, the shaft carrying with it the ratchet wheel K will simply run ahead of the slower rotating gear carrying the pawl. When out of use the pawl is held away from the teeth by means of a spring snapped into a groove in the ratchet wheel, the ends of the spring abutting against a projecting piece or wing on the side of the pawl.

There is a repetition of this arrangement in the other set of gears Z, and as there are three changes in each set, a total of nine is given out by the shaft L, which may be connected to the main gearing of the machine by suitable means. It will be noticed that there is no out of gear position. When the two levers are upright as shown, both the ratchet wheels are driving, and the shaft L is running at its slowest speed. Fast and loose pulleys are provided so that the motion of the gears may be stopped. The arm carrying the strap bar may be set over into any position to suit the inclination of the belt, and a

half-turn of the handle M throws the belt on or off. The gears run in a bath of oil, and the shaft journals are provided with ring oilers, and arranged so that no oil shall run outside. The changes being made by tooth clutches the drive is positive and the box a very powerful one. As the gearing and shafts are constantly in motion there is hardly any shock when changing speeds, even when running at the high speeds. For lower speeds of the pulleys than that

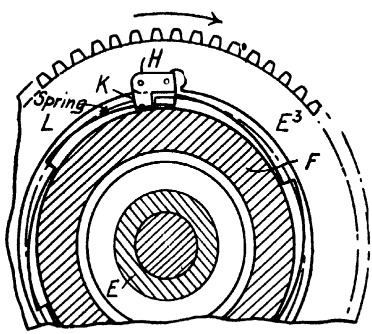


FIG. 31.—Silent pawl wheel.

given there is no shock whatever, and with the high speeds this is avoided by the simple act of throwing the belt off and on again.

Fig. 31 shows an enlarged view of a ratchet wheel and silent pawl from the practice of the Jones & Lamson Company. It is termed "silent" on account of its being held out of gear with the ratchet wheel when running freely and so avoiding any unpleasant clicking. The pawl H is dropped flush into a pocket in the face of the gear. On the outside of

the pawl is riveted a wing K, and in a groove in gear G is snapped a spring L (shown in long and short dotted lines), the ends of which abut against the wing K. When the ratchet wheel rotates faster than gear E^3 which carries the pawl H, the pawl itself is held away from the ratchet wheel by the action of the spring bearing against the wing K. Again, when gear E^3 carrying the pawl rotates faster than the ratchet wheel F, the pawl is turned into mesh with F by the push of the spring on the opposite side of the wing.

The head shown in Fig. 32 is a very interesting example of the use of friction clutches in conjunction with the silent pawl. It is the cross sliding head of the Jones & Lamson flat turret lathe.

There is only one gear on the spindle, but that is a double wheel U, made fast in the manner indicated, and between this gear and a thrust web cast in the head are two hardened and ground steel washers forming a thrust bearing. At the front end of this gear is a collar that is grooved across one face to receive the ends of two pins which are carried by lugs in the casting, and which may be adjusted against the collar by pointed setscrews so as to take up all end movement of the spindle.

The driving pulley is at the front, and is keyed to a shaft D, to which are pinned three small gears, D^1, D^2, D^3, which are in mesh with three mating gears E^1, E^2, E^3 on shaft E. Gear E^3 is chambered out at one side to receive a ratchet wheel F, which is formed integrally with a spur gear G keyed to shaft E; a silent pawl H forms, in conjunction with the ratchet wheel, a connecting medium be-

tween gears E^3 and G, and thus shaft E may be driven through gears D^3 and E^3 at a constant slow rate of speed. Between gears E^1 and E^2 is keyed a sliding double friction cone J, which may be moved to right or left to clutch either gear to the shaft E. This shaft then has three rates of speed, and so long as the driving shaft D is running, will rotate at its slowest speed if the friction clutch J is left in mid-position. If either friction gear is engaged, however, the shaft carrying with it gear G and ratchet wheel F will, of course, simply run ahead of the slowly rotating gear E^3 carrying the pawl H.

The mechanism for moving the sliding clutch is a very neat one, and will be easily understood by reference to the illustration, and also to Fig. 15. Use is made of the little toggle levers that have proved so effective in the automatic chuck on this lathe. In shaft E is fitted a sliding rod M, which at its inner end carries a pin passing through a slot in the shaft, and connects the sliding clutch J to the rod. The rod itself has a hole down at its outer end, and in this hole is placed a second rod N, which is cut away so as to make two inclined surfaces on opposite sides. These inclined surfaces, or wedges, operate the little cams N^1 and N^2, which are pivoted and rest against the sides of the adjustable collar N^3 as shown. When rod N is pushed inward by moving the handle to the right, N^1 is lifted, and the cam surface at its end bears against the bottom of the slot cut through rod M, and causes M to be forced to the left, engaging friction clutch J with E′. When the operating handle is thrown to the left, as shown, cam N^2 is swung out, drawing

M to the right, and causing gear E^2 to be clutched to its shaft. The other friction clutches are all operated in the same manner; the lever O controls the friction clutch Q on shaft P, which starts, stops, and reverses the motion of the spindle, and is operated by a long shipper rod located above the machine. To keep the friction clutch gears apart, a series of spacing pins is fitted to slide freely in the hubs of the sliding cones, two of which may be seen between the hubs of each pair of gears.

The drive to shaft P is obtained through three sets of gears R, R^1, and R^2 on shaft P, and their corresponding gears on shaft E. Gear R, however, is driven from pinion R^3 through a compound gear placed above it and running freely on a stud, not shown in the illustration. The pinion of the compound gear meshes with gear R, and this gear gives the slowest speed of the combination. Gear R carries a ratchet wheel similar to E^3.

It will be seen that this arrangement of gearing gives for each speed of shaft E three rates of speed for shaft P, or a total of nine speeds and operated by two levers. These are transferred to the spindle through the medium of the two friction gears S and T and the large double wheel U, gear S driving the spindle forward, and gear T, through an intermediate pinion below, giving it a reverse motion. With the cone Q in mid-position the spindle is stationary.

Another head used on a turret lathe by the Greaves-Klusman Tool Co., U.S.A., and embodying the friction clutch and silent pawl combination, is shown in Fig. 33. The driving pulley for this head is provided with a friction clutch for starting and stopping,

this being of the expanding ring type. The other

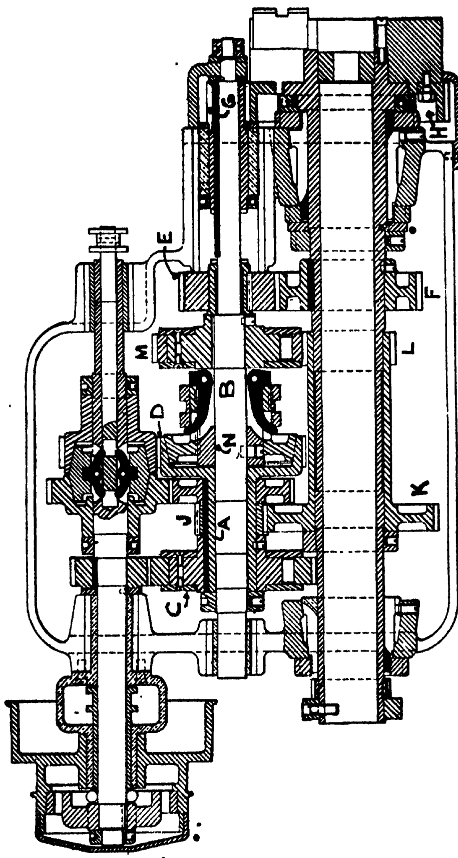


FIG. 33.—Greaves-Klusman all-gear head.

clutches for speed changing are conical, and by an

ingenious arrangement of the gears twelve speeds are given to the spindle, being through either high or low back gear, ranging from 7 to 230 R.P.M. when the driving pulley makes 350 R.P.M.

On the driving shaft the sliding cone is operated by the rod inside the shaft acting by endwise movement against the ends of the double levers pivoted in the shaft, two speeds being thus transmitted to a sleeve A running freely on shaft B, and a third through ratchet gear C. From sleeve A motion is transmitted through either single or treble gear. To operate the single gear the cone clutch in gear D is engaged by the sliding collar and levers, and drives the piece N which carries the levers and is attached to the shaft B. The motion is then through either gears E and F or G and H, the latter being attached direct to the face-plate and giving a very powerful drive. Thus six speeds are given to the spindle, and are the quickest running of the set. The remaining six speeds are obtained by the use of a silent pawl in gear M, the cone clutch in D being disengaged, when the constantly running double gear J, K, L, and M comes into action and drives the shaft B. On the quicker set of speeds, the ratchet wheel of course simply runs ahead of its gear as already explained.

The silent pawl system is also used in a novel arrangement of driving for a car wheel boring machine. A two-speed counter-shaft is used, and this, in combination with the three-speed mechanism incorporated in the machine, gives a range of six changes without clutches of any kind. As shown by Fig. 34 the change in the speed is obtained by shifting the single driving belt on to either one of the three driving

pulleys provided. When the belt is running on the central pulley A the drive is direct. Pulleys B and C are mounted on sleeves running freely on the driving shaft D. These sleeves have teeth cut out on their outer ends to form pinions meshing with gears E and F; these latter, in turn, through gears and the shafts on which they are mounted drive the ratchet members G and H. Taking that at the left-hand side, M is the ratchet wheel which is keyed to shaft D, J is the pawl carried by the gear G, K is a wing brazed to J, and pressed against the face of M by means of the spring L. Consequently, when the belt is shifted on to pulley B, and with the gear G travelling in the direction indicated by the arrow, the friction of the wing K against the ratchet gear M draws the pawl down into the position shown and into engagement with the ratchet teeth, the shaft D then being driven at the speed of the gear G. When the shaft is driven by pulley A, the ratchet gear of course rotates faster than gear G, and the action of the wing is to lift the pawl clear of the teeth and hold it out of the way until the slow gear again becomes the driver for the shaft. Similarly, if the belt be shifted to pulley C the shaft is then driven by the gearing at the right hand. In either case, only that train of gearing is in motion which is directly driven by the belt.

An eight-speed clutch gear box by Messrs. Jones & Shipman, Ltd., Leicester, as used for a drilling machine, is shown in Fig. 35. It will be seen that there are twelve gears, and three handles are necessary to move the clutches. The gear runs are as follows:—

C, D, G, H,	Spindle	speed	400	R.P.M.
C, D, E, F,	"	"	340	"
J, K, G, H,	"	"	289	"
J, K, E, F,	"	"	245	"
M, N, O, P, C, D, G, H,	"	"	208	"
M, N, O, P, C, D, E, F,	"	"	176	"
M, N, O, P, J, K, G, H,	"	"	150	"
M, N, O, P, J, K, E, F,	"	"	127	"

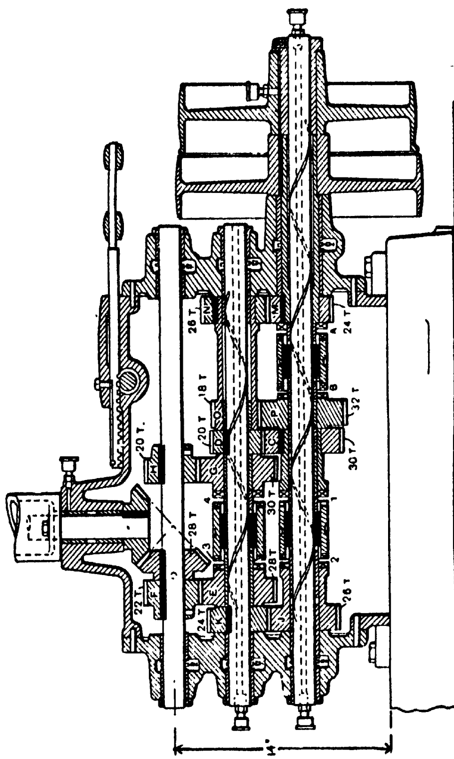


Fig. 35.—Jones & Shipman eight-speed gear box.

Toothed clutches lend themselves to a variety of adaptations. For instance, in Fig. 36 we have what is termed a "triple" clutch, i.e. three speeds may be imparted through the medium of the sliding member keyed to the shaft. A, B, and C are gears which receive motion at three different speeds, and run freely on the shaft. A and C are provided with

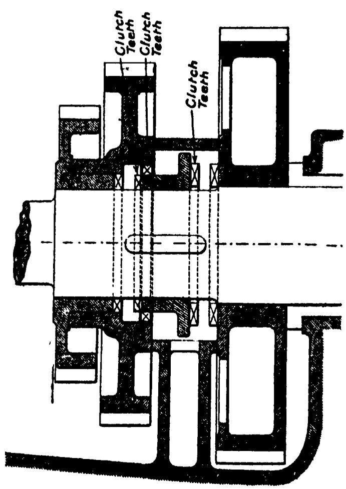


FIG. 36.—Triple clutch.

clutch teeth of the ordinary kind on their faces, and the sliding member D has corresponding teeth. But the teeth for the middle gear B are formed on the periphery of D, and on the inside of the bore of the gear. As shown, these teeth are in mesh, and B is driving the shaft, A and C running idle. To engage either A or C, the peripheral teeth must pass

clear through the teeth of B. There is an out-of-gear position for each. A ring on D is the means provided for moving it instead of the groove as in the usual form of clutch. It will be obvious that another external clutch could be arranged on the right-hand side, and with the addition of another gear be made into a four-speed clutch.

Sliding-gear Type.—Leaving the clutch examples, we will now turn our attention to those in which speed changes are made by sliding gears. A four-speed box is shown in Fig. 37, attached to the side of a pillar-type shaping machine. As eight speeds are required a double gear is introduced inside the column, so doubling the number of changes given by the box. In this example the box is not a separate unit, as the driving shaft runs right through the machine, the driving pulley being on the left-hand side. An expanding ring clutch—the outside of which is acted on to form a brake—is fitted on the right-hand side, the body being formed into a sleeve surrounding the shaft. The four gears mounted on the sleeve may be engaged by the four sliding gears mounted on the shaft above. It will be seen that every pair of gears has an out-of-gear position, and only the gears actually required are in mesh. This is a most valuable feature and one in which the sliding-gear system predominates.

Changes can be made instantly without stopping the belt. When the clutch is out the pulley shaft alone is running, and after the gears are disengaged the sleeve creeps slowly around without force, making it easy to engage gears absolutely without shock, the

best results being obtained by handling the shifter lever as rapidly as possible.

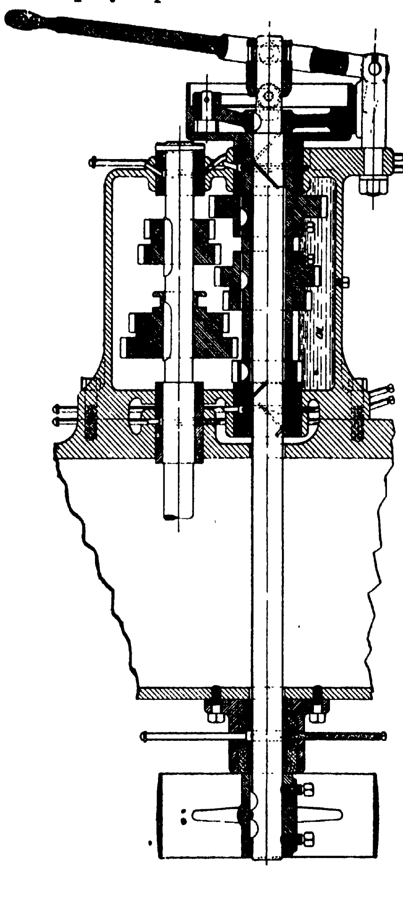


FIG. 37.—Four-speed box used on the Queen City shaper.

The gear-shifter handle has a universal joint to allow the four necessary movements; a device as shown in Fig. 38 provides for non-interference of gears, it being impossible to engage more than one pair at the same time, and the control is "selective," the movement being to and from the neutral point where all gears are disengaged.

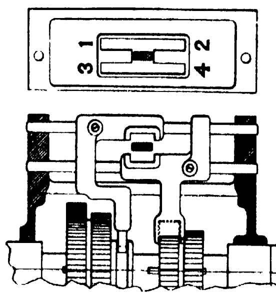


FIG. 38.—Gate change speed arrangement.

The four gear box changes can be made in as many seconds, and the gears are each time securely locked in their respective idle and working positions by automatic spring plungers. Numbers 1, 2, 3, and 4 on the cover plainly indicate where to throw the lever to secure the speeds from low to high respectively. This method of moving sliding gears is known as the "gate" change, and is further exemplified in Figs. 39 and 40, a double gate being used for obtaining sixteen changes of speed.

The sixteen speeds range in geometrical progression from 200 to 10 R.P.M., and all these changes are obtained by the operation of but two handles. Whatever combination is in use, *there are only three*

pairs of gears in mesh at any one time, and in no case are there any idle gears in mesh. Further, there are no loose sleeves running, all gears being keyed to the shafts. The driving pulley runs at 480 revs. per min. It is 18 ins. diam., and

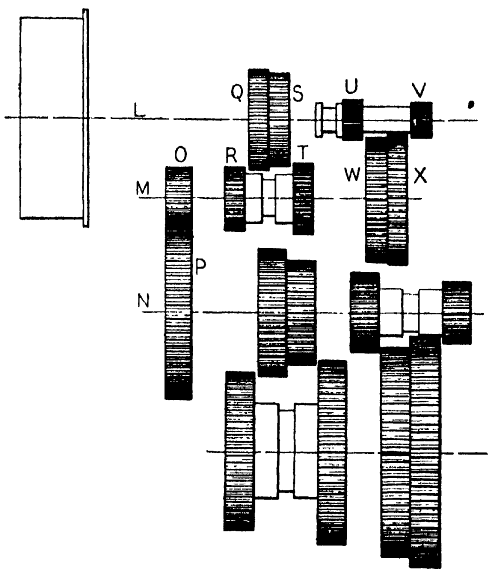


FIG. 39.—Lay-out of sliding gears to give sixteen changes.

driven by a 6-in. belt. It runs loose on a phosphor-bronze bush, so that there is no strain on the shaft due to the pull of the heavy belt. It may be connected to the driving-shaft by means of a powerful friction clutch of the expanding ring type, and the arrangement is such that when the clutch is thrown

out a slight further movement of the handle brings into operation a brake which, acting on the side of the gear Q, quickly brings the motion of all the gearing to a standstill.

The position of the handles to give any desired speed is obtained from a speed chart given below. There are two distinct sets of gearing, and each handle operates its own particular set. In each set the arrangement of the gears is alike, and in each case the gears are in pairs, whether fixed or sliding. Two pairs of gears in either set cannot be in mesh

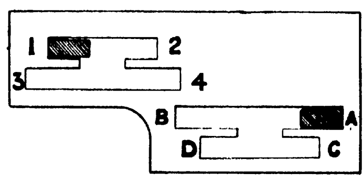


Fig. 40.

at one and the same time, and when the two levers are in the middle position all the gears are out of mesh. The two sets are connected by a pair of gears O and P.

The movements of the respective gears to form the different combinations are effected as follows: Assume Q and R to be in mesh. Gear Q is fixed on the shaft, and R is a sliding gear. By throwing the operating handle into position 2 (Fig. 40), this gear may be moved along its shaft, and gears S and T brought into mesh. As these are of a different diameter a change of speed is thus given to the spindle. The next change is effected by first throw-

ing the double sliding gear R, T, into the mid-position, so that both pairs of gears are out of mesh, and the operating handle is vertical. This handle is then opposite to the centre opening in the I bracket, which guides it and determines its position. It is pulled through this opening, and is then free to work in the front portion of the slot. By this movement the handle is disconnected from the double gear R, T, and, instead, is connected with the double sliding gear U, V, which may then be caused to mesh with either gears W or X. Thus four changes of speed are given to the second motion shaft M, and thence transmitted at a reduced speed through the fixed gears O and P to shaft N. Then, as there is a second set of four pairs of gears, for each of these latter pairs there are four speeds, making a total of sixteen speeds.

The connection of the operating handles to the two pairs of gears is as follows: The shaft, on which the handle may work freely, passes through a hollow shaft to the rear of the headstock. Both the solid and hollow shafts have, at the front side, large collars of equal diameter and width, in which two grooves are cut across. The handle is made with two projecting claws which are shaped to fit these grooves. When in the rear slot 1, 2, the claws fit the grooves in the hollow shaft, and this then moves with the lever, the solid shaft being held stationary by a spring plunger. By placing the handle in the middle position the grooves in the two collars come opposite to each other, and by drawing the handle into the slot 3, 4, the claws are drawn from the hollow into the solid shaft, which then moves with the handle,

while the hollow shaft remains fixed. From the two shafts the motion is given to the respective sliding gears by a suitable arrangement of sliding racks and forks. Both sets of gearing are operated in the same way.

The chart shows the position of the handles to obtain the various speeds. The changes of speeds can be made very readily. By suitably operating the friction clutch by means of its handle, in combination with the action of the brake, the motion of the gearing may be brought almost to rest and the changes effected without actually stopping the head.

REVOLUTIONS OF SPINDLE PER MINUTE.

Positions of Handles.	Revs. per min.	Positions of Handles.	Revs. per min.	Positions of Handles.	Revs. per min.	Positions of Handles.	Revs. per min.
1A	200	2A	160	1B	136	2B	109
1C	91	2C	73	1D	62	2D	50
3A	39	4A	32	3B	26·4	4B	22
3C	17·8	4C	14·6	3D	12·2	4D	10

Fig. 41 shows a modification of the last arrangement, and how sixteen changes can be obtained by the use of sixteen gears. In this case the gears on the centre shaft are fixed, and any one of the four pairs may be moved as required to affect the combinations. With a suitable arrangement, such as the gate control described, two handles may be used to produce all the combinations of speeds. With this arrangement of gears, however, there is one serious difficulty. It will be noticed that the total number of teeth in any one pair on the first two shafts must equal the total number of teeth in any other pair on

those shafts, and likewise with the second two shafts. From this it will be readily seen that a change of one tooth in any one gear affects several other gears, and therefore changes slightly almost all the ratios in the combination. For this reason a

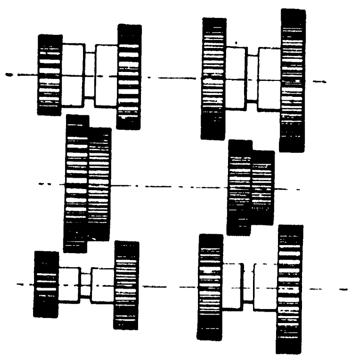


FIG. 41.—Another method of obtaining sixteen changes with sliding gears.

correct geometrical progression cannot always be obtained, and the arrangement shown in Fig. 39 is preferred.

A cam is frequently used to make the changes in sliding gears. By its use one pair of gears may be moved at a time, and it thus forms a simple method of ensuring that two pairs of gears on the same shaft are not in mesh at the same time. In Fig. 42 is shown a sixteen-change speed box, fourteen gears being required, and three operating handles, one of which turns a cam (Fig. 43) contained in a recess in the side of the box. The photograph (Fig. 42) shows

the gears in mesh to give the second slowest speed. The drive is through A, B, the latter running freely on the shaft, but keyed to a sleeve carrying three

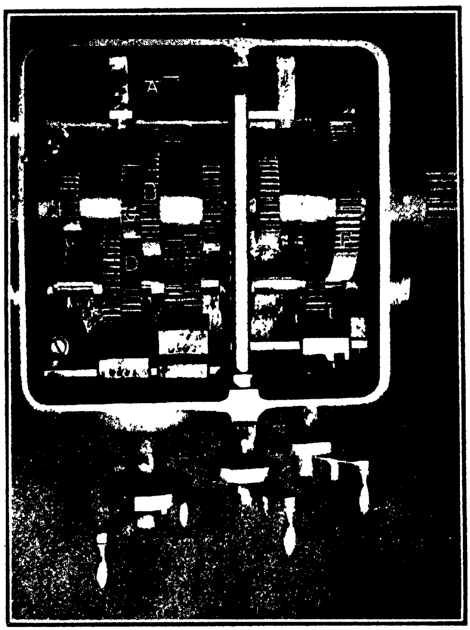


FIG. 42.—Cam-operated gear box.

other gears, and C, D; then through the shaft and finally E, F. Gear F is keyed to a shaft on the end of which a pinion is formed integrally, and transmits the motion from the box as required.

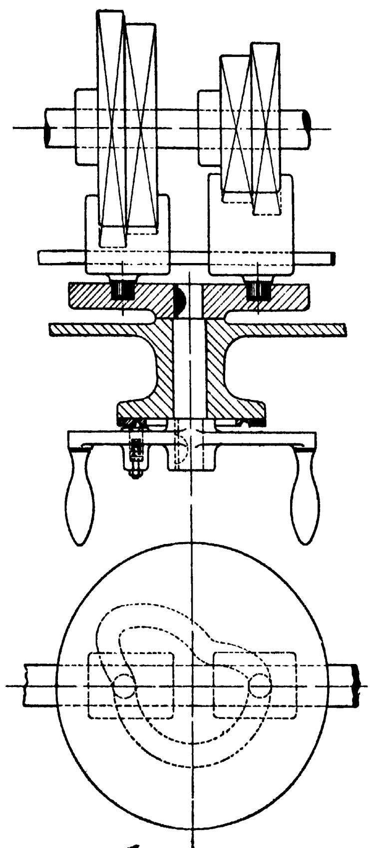


FIG. 43.—Details of cam in Fig. 42.

It is always advisable to have an interlocking motion when two pairs of gears are carried on one shaft, so as to ensure there will be no chance of both sets of gears ever being in mesh together. An ingenious device is shown in Fig. 44. Two similar quadrants A and B are attached to the handles for moving the gears. On the outside of each, where

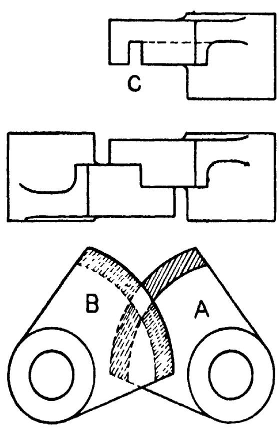


FIG. 44.—Interlocking device.

the quadrants intersect each other, the rim is cut away as shown at C. In the position shown the two handles would be in the middle position and the gears disengaged, and the open space would allow one of the handles, say at A, to be moved, the shaded rim of the quadrant A passing into the groove in the other, B. Quadrant B would thus be held by the rim of A, and prevented from movement until A had been returned to the middle position.

An **outer bearing** is often used to support the end of the driving shaft and pulleys. This is a very good feature, because with heavy belts and a large amount

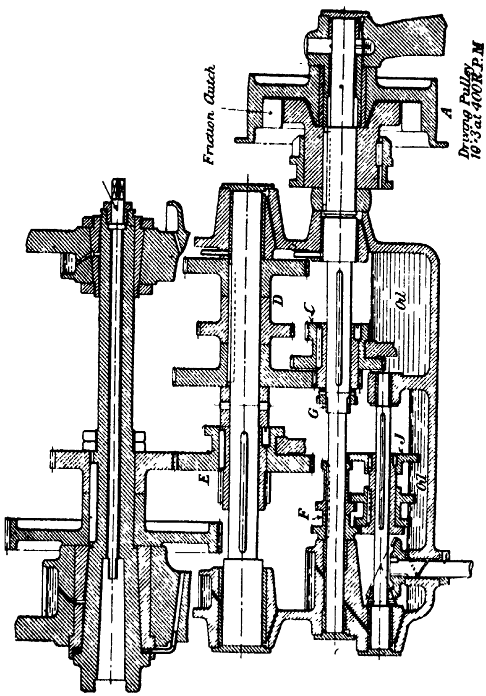


FIG. 45.—Archdale's milling machine drive.

of power as usually transmitted by an all-gear head, the driving shaft is naturally subjected to a heavy belt pull and a tendency to spring. To offset this tendency the outer end of the shaft is mounted in a long bearing in a support carried on the head.

This outer bearing is in evidence in Fig. 45, the drive for a *horizontal milling machine* by Messrs. Jas. Archdale & Co., Ltd., Birmingham. In the one head are combined both the spindle change mechanism and the feed change, all by sliding gears, there being six speeds to the spindle and four feeds. In this case the driving pulley A runs freely on a boss on the outer bearing so that no belt pull is transmitted to the shaft, an expanding ring friction clutch being used to connect with the shaft. To change the speeds the nest of three gears C is moved to bring any one of them into mesh with their mating gears in the nest at D. Note that with this construction the centre gear on D must always be smaller than either of the other gears in order to allow the gears on C to slide past, otherwise more space is required as shown at F. The motion is then transmitted through another pair of gears E direct to the spindle.

The **feed gears** are driven from the driving shaft. Three of these are in a nest at F, and the fourth one G has its teeth cut in the driving shaft. By the movement of one handle all the four gears in the nest at J are moved bodily to engage with their respective gears. Where one handle is used with four gears it is, of course, necessary when changing speeds to slide gears through one another. Where two handles are used, and an interlocking device, the gears do not slide through, only in and out.

In order to effectively lubricate the whole system of gears and bearings, a common supply of oil is pumped up from a tank in the body of the machine, comes down through pipes at the top, and floods all the bearings and gears. The various pockets and

channels to the bearings, the drains and protective caps on the ends are all clearly shown.

A type of all-gear head used by Messrs. Tangyes Ltd., of Birmingham, and recommended for lathes above 12-in. centres, is shown in Figs. 46 and 47.

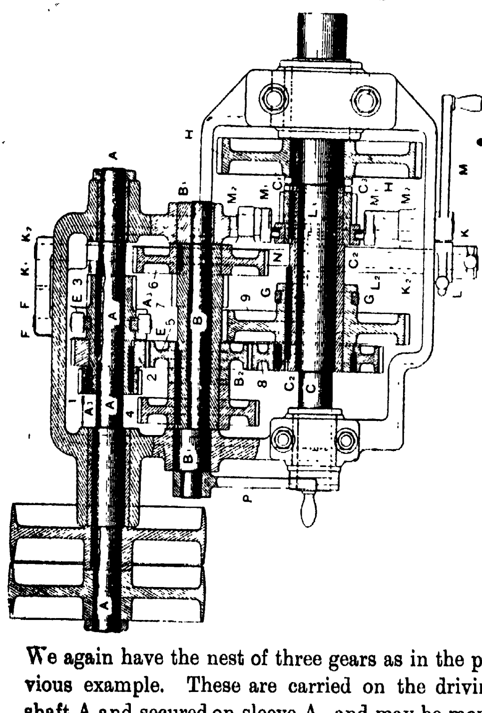


FIG. 46.—Tangye's headstock. Sliding gears and clutches.

We again have the nest of three gears as in the previous example. These are carried on the driving-shaft A and secured on sleeve A_3, and may be moved to mesh with their corresponding gears 4-5-6 on shaft B, by means of the handle K in front of the

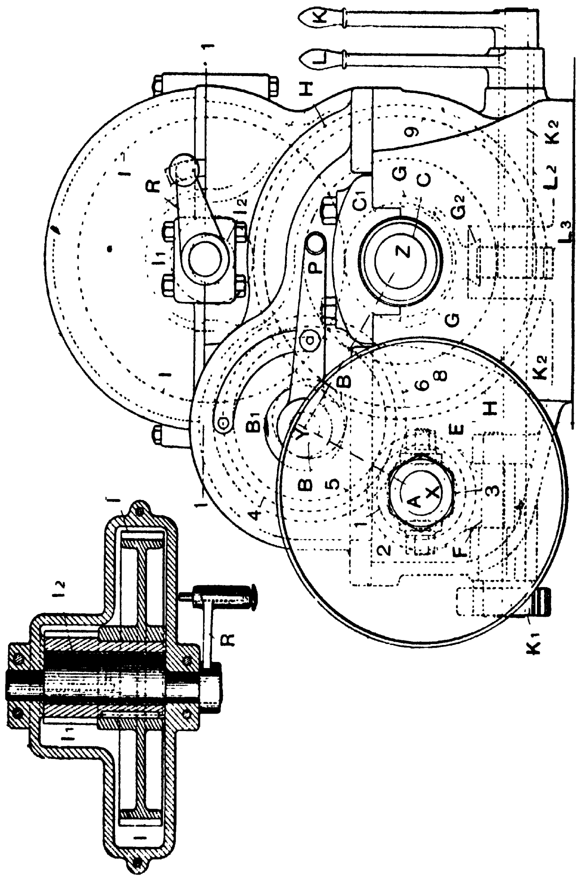


FIG. 47.—End view of Tangye's head.

head. On the spindle C is a sleeve C_2, and to the spindle is keyed a gear H. On sleeve C_2 are keyed two gears 8 and 9 and a pinion C_1. Gear 8 can mesh as shown with 5 on shaft B, and by sliding this gear along the sleeve by means of the handle L, 9 may mesh with 7, and consequently six changes be given to sleeve C_2. In addition, there is a double-gear arrangement carried above, so that this number of speeds is doubled and twelve changes obtained.

To permit of the free endwise movement of the sleeve A^3 or connected gears 8 and 9, when it is desired to effect changes in the speed of the spindle, as well as to prevent injury to the teeth of the gears on making the changes, provision is made for the raising of the intermediate shaft B to a height sufficient to take the gears thereon out of mesh with the gears on the shafts A and C. For this purpose shaft B is provided with eccentric bushes, B, B_1, and may be rotated by means of the handle P. Similarly the eccentric shaft L_2 carrying the "back gear" may be raised by turning the handle R (Fig. 47) when the pinion C_1 is to be thrown into direct engagement with gear H by the engagement of the clutches C_3. The endwise movement of the connected gears 8 and 9 is effected by the movement of the fork G, which moves in a dovetail slide and carries a rack which is in mesh with a pinion on the hollow shaft L_2.

The *ratios of the gears* are as follows:—

Without back gear: 2, 2·9, 4·15, 6·36, 9·22, and 13·2 to 1.

With back gear: 20·37, 29·53, 42·26, 64·16, 93·7, and 134·4 to 1.

Castellated Shafts.—For sliding-gear quick-

change speed boxes circular shafts with inserted keys have serious defects. A shaft having a key laid in one side is considerably weakened by the keyway, and the key loosens under the repeated shocks of use. This is overcome to a certain extent by the use of keys deep in the shaft, or pinned, or held by set-screws. Also when shifting a gear or a clutch under load, twice the effort is required to move it along the

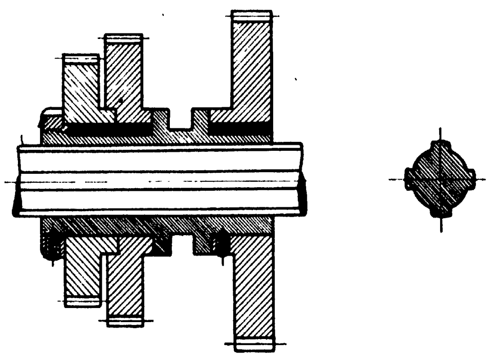


Fig. 48.—Castellated shaft.

shaft than when the gear is driven by two keys diametrically opposite; but two keys located opposite do not do away with the trouble of loosening.

This weakness of the inserted key suggested the old-time square shaft, which having four solid faces, or keys, allows a gear to be moved very freely on it. The square shaft, however, does not receive much favour because of difficulties of manufacture, and it is much heavier than it need be for a given torsional strength. The integral splined shaft, or castellated shaft, was therefore developed. This is a round

shaft in which the keys are produced by milling out the metal between, thus producing four, six, or eight keys integral with the shaft. Fig. 48 shows a nest of gears with sleeve on a four-keyed shaft, the most usual type. The advantages of this type of shaft are so evident that wherever the sliding-gear construction is used, it should be incorporated in the design.

Tumbler-gear Type.—The tumbler-gear type of speed change is one that is very popular. In combination with sliding gears it is possible to obtain the greatest number of speed changes with the lowest number of gears. It is so called because it consists of a nest of gears keyed together on one shaft, and driven from another shaft by means of an intermediate gear (or tumbler gear) which may be caused to mesh with any one of the gears in the nest, and still remain in mesh with the first driving pinion. As may be expected, there are many and various arrangements for carrying and operating the tumbler.

The gear box used on the Cincinnati Bickford drill is shown in Fig. 49 and the nest of gears detached in Fig. 50. There are eight speeds obtained in the box, afterwards trebled by gears on the spindle head, making twenty-four speeds in all. The driving shaft A is constantly running, and the tumbler bracket is pivoted about this shaft. There are seven gears in the nest C, and an additional set of gears at D, the slowest running of the combination, has a silent pawl device, so that shock is avoided when engaging gears, because the ratchet gear prevents the speed of the driven shaft from falling below the minimum speed. The speed-plate is cast in bronze, and when

in position hides the set-screws which regulate the

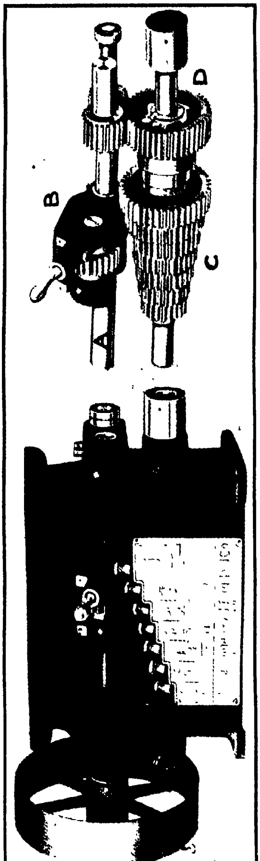


Fig. 49. Cincinnati Bickford tumbler gear box. Fig. 50.

depth of engagement of the tumbler gear, and shows,

by direct reading, where to place the levers in order to obtain a cutting speed of 35 and 70 ft. per min. With these points to serve as a guide the setting for any other rate of speed is easily determined.

As showing the great number of changes possible with the tumbler gear we may take the example given in Fig. 51, where it is possible to obtain twenty-one changes in geometrical progression with only twelve gears and four shafts. The driving pinion is A, and the

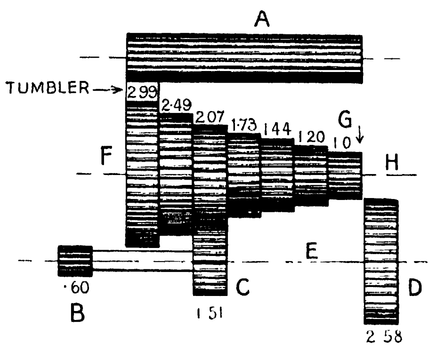


FIG. 51.—Diagram of tumbler gear arrangement.

tumbler gear is always in mesh with it. This may engage with any of the gears in the nest. Gears B, C, and D are keyed to the driven shaft E, and may be engaged as shown, or C may be moved out of mesh, and B allowed to mesh with the largest gear of the nest, F; or D may be engaged with the smallest gear G. The shaft H has thus seven speeds given to it, corresponding with the number of gears in the nest, and for each of these speeds shaft E has three, making twenty-one in all.

The relative sizes of the gears are:—

1 - 1·20 - 1·44 - 1·73 - 2·07 - 2·49 - 2·99.

The pinion A would be 1.

It will be noticed that these are in geometrical progression with a rise in the ratio of 1·20 to 1. On looking at our tables on pages 14 and 15, we can read off opposite ratio 1·20 the full set of speeds obtainable, the total ratio between highest and lowest being found under column 21 as 38·34 to 1. Again examining the table, under column 8 will be found the first double-gear ratio, viz. 3·58 to 1; and under column 15 the second double-gear ratio, viz. 12·84 to 1.

To obtain the sizes of gears on shaft E we proceed as follows:—

A ratio of gear is already existing in the nest of 2·99 to 1;

the ratio required as shown by table = 3·58,

$\therefore \frac{3 \cdot 58}{2 \cdot 99} = 1 \cdot 2$, which as C is in mesh with a gear of the ratio of 2·07, gives $2 \cdot 07 \times 1 \cdot 2 = 2 \cdot 49$ as the ratio for C.

In like manner the other ratio can be found.

The ratio existing in the nest = 2·99 to 1, the ratio required as shown by table = 12·84.

$\therefore \frac{12 \cdot 84}{2 \cdot 99} = 4 \cdot 3$, and as D may mesh with gear of 1, the ratio of 4·3 remains.

Now for the gears to mesh, it is essential that the sum of each of the ratios has the same total, and it will be quite obvious that the ratios found will not do so. Therefore it will be necessary to multiply all the ratios by a common multiplier in order to obtain uniform total ratios, in this case it is ·6.

Gear B thus becomes ·6 diameter,
C becomes 2·49 × ·6 = 1·51 diameter,
and D becomes 4·3 × ·6 = 2·58 diameter.

The sums of the ratios are:—

2·99	2·07	1·00
·60	1·51	2·58
3·59	3·58	3·58

Note.—The varying figures are similar to those in the table.

The Effect of Oil Depth on Efficiency.—A gear box should not have an excessive supply of oil, and the exact depth intended should always be shown by some form of indicator to register the level of the oil. A drain is also necessary, so that the oil may be renewed from time to time. Tests on machine tools have shown that the power needed increased uniformly as the oil level was raised. There is no doubt the extra power was entailed by the additional work to be done in moving the oil, all the gears acting as if they were pumps, and taking the power that a pump would require to do the same amount of work.

In 1915 the National Physical Laboratory made some tests on the gear box of an automobile for the influence of oil depth on efficiency, and these tests also showed that for high efficiency the oil must be kept relatively low. With an oil depth of one-quarter the box depth the efficiency was 97·5 per cent; at half-depth 94 per cent; at three-quarters 90 per cent; and this further fell to the astonishingly low figure of 74 per cent when the box was full of oil. These results were obtained on top gear. They were not materially affected, however, by running on

the second or third gear, but of course the viscosity of different oils had an influence.

Summary of Drives.—In Figs. 52 to 64 are given

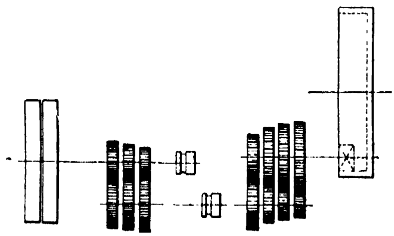


FIG. 52.

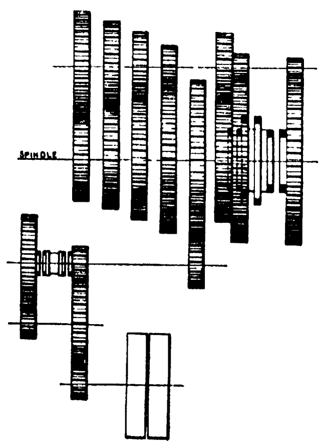


FIG. 53.

FIGS. 52 and 53.—Sliding-key types.

in diagram form a series of illustrations of various all-gear drives.

Fig. 52 shows a sliding-key operated mechanism, 14 gears being required to give 12 speeds.

Fig. 53. Sliding-key and clutch combination—21 gears giving 24 changes of speed.

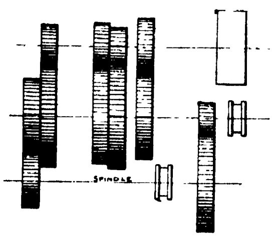


FIG. 54.

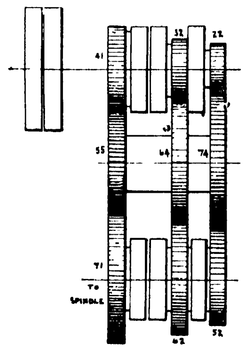


FIG. 55.

Fig. 54. Friction-clutch system—12 gears, 8 speeds.
Fig. 55. Clutch and ratchet system—9 gears, 9 speeds.

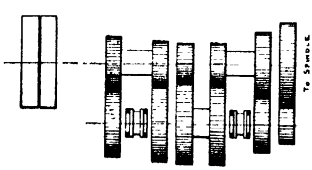


FIG. 56.

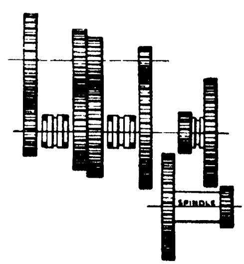


FIG. 57.

Fig. 56. Clutch and ratchet system— 9 gears, 9 speeds.

Fig. 57. Clutch and sliding-gear system—12 gears, 8 speeds.

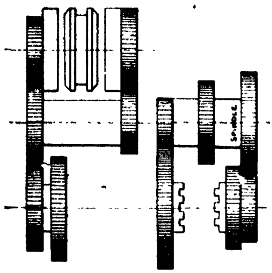


FIG. 58.

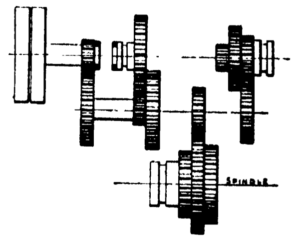


FIG. 59.

Fig. 58. Clutch and sliding-gear system—12 gears, 12 speeds.

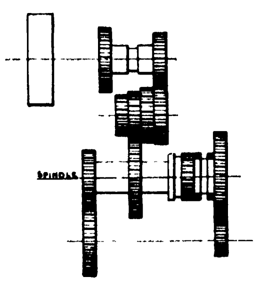


FIG. 60.

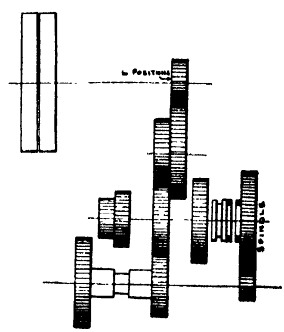


FIG. 61.

Fig. 59. Clutch and sliding-gear system—13 gears, 18 speeds.

Fig. 60. Tumbler-gear system—4 gears carried in tumbler frame, 11 gears, 16 speeds.

Fig. 61. Tumbler-gear system—2 gears carried in tumbler frame, 11 gears, 18 speeds.

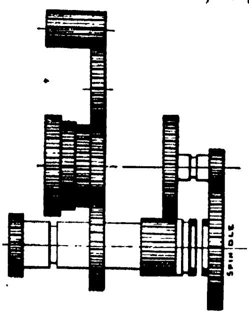


FIG. 62.

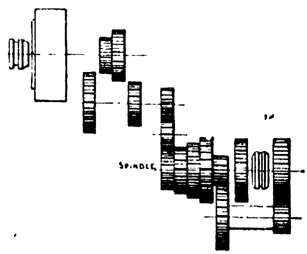


FIG. 63.

Fig. 62. Tumbler-gear system—12 gears, 16 speeds.

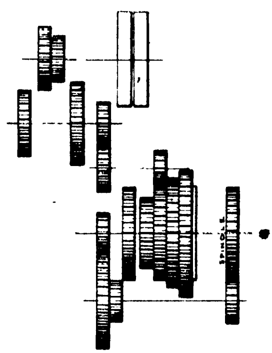


FIG. 64.

Fig. 63. Tumbler-gear system—15 gears, 20 speeds.

Fig. 64. Tumbler-gear system—17 gears, 24 speeds.

CHAPTER V.

APPLICATIONS OF MOTOR DRIVE.

Advantages of Individual Drive.—In cases where it is possible to use motors for the individual driving of machine tools, the following advantages accrue due to the elimination of belts and shafting. The absence of belting tends to increase cleanliness and gives unobstructed light, both natural and artificial, and the space overhead being left clear, more extended use can be made of overhead cranes, with the consequent saving of time and expense in handling materials. Also in shops where the individual motor drive has been introduced, the saving of power previously wasted in turning line shafts and belts has resulted in big reductions in power bills. A complicated system of belts and line shafts is an expensive thing to keep in good working order, requiring constant attention; the modern electrical motor, on the other hand, requires but little beyond periodical oiling and cleaning, so that the maintenance expense is practically negligible.

Further, the individual drive allows a machine to be placed anywhere in the shop where most convenient, independent of any line shaft, and certain types of machines such as drills, shapers, and slotters,

may be moved about by the crane and placed in position alongside the work, a great convenience when this happens to be of large dimensions.

Types of Motors.—In any machine shop there are distinct conditions of operation calling for different methods of applying motors to machine tools. Where direct current is available there are three types of motors which may be used, the proper type being selected for the particular machine to which the motor is to be applied. These three types are series-wound, shunt-wound, and compound-wound motors.

A Series-wound Motor is one in which the field winding is in series with, or forms a direct continuation of, the armature circuit, so that all the current that passes through the armature passes also through the fields. The amount of current flowing through a motor is dependent upon the power the motor is developing, and it therefore follows that in the series motor the strength of the field will fluctuate with the load placed on the motor. As the speed of the motor depends inversely upon the field strength, the speed of the series motor will be inversely proportional to the load. The characteristics of the series motor are, therefore, heavy starting torque and speed dependent upon the load.

The Shunt-wound Motor is one in which the field winding is connected across the mains, or is said to be in shunt with the armature circuit. The amount of current passing through the fields is inversely proportional to their resistance, and remains practically constant under all conditions of load. This results in an instant speed motor whose

output is dependent upon the amount of current which passes through the armature. The characteristic of the shunt-wound motor is, therefore, approximately constant speed under all conditions of load.

A Compound-wound Motor is one having both a shunt and a series-field winding. The shunt field is connected across the main lines, as in a shunt motor, while the series field is in series with the armature and carries all the current passing through it, as in the series motor. The speed of a compound-wound motor is more nearly constant than that of a series motor, but the dropping speed from no-load to full-load is considerably more than in a shunt-wound motor, owing to the action of the series winding. The characteristics of the compound-wound motor partake of those of both the series and shunt motors in about the same degree of the relative proportion of the two windings composing the field.

With direct current available we may have either constant-speed or variable-speed motors. On the other hand, the alternating current motor is essentially a constant-speed machine. At the present time no commercial method has been found for varying the speed of an alternating current motor in such a way that it can be successfully used for machine tool work, so that with such a motor it is absolutely essential to employ a gear box to obtain the various speeds required.

Selection of Motors.—To determine the type of motor to be employed for the different classes of machine tools, the character of the power requirements should be considered. Where the work of

cutting is continuous, and no excess of power is needed to start the machine, the shunt-wound motor should invariably be used. In the case of planers, shapers, slotters, etc., the work is intermittent, being far greater in some portions of the stroke than at others, and for this class of work the compound-wound motor is best suited. The following table will form a useful guide :—

Types of Motors for Driving Machine Tools.

Lathes . . . . . . Boring and Turning Mills . Radial Drills	Variable Speed Shunt-Wound Motor.
Milling Machines . . Grinding ,, . . Cold Sawing ,, . . Band ,, ,, . . Wood Planing ,, . .	Constant Speed Shunt-Wound Motor.
Planing Machines . . . Shaping ,, ... Slotting ,, ... Punching & Shearing Machines Plate Bending Machines . .	Compound-Wound Motor with Fly-wheel.
Auxiliary requirements . . Raising Cross-Slides . . Traversing Heavy Tables, Carriages, etc. . . .	Heavy Starting Torque. Series-Wound Motor.

An analysis to determine exactly the length of time a machine tool can be expected to be in operation, and which took into account the time of loading, cutting, unloading, and other details occasioned by miscellaneous causes, showed conclusively that it was not necessary to use a continuously-rated motor.

In fact, an intermittent rating of the motor for a period not exceeding two hours' continuous service answers for all kinds of machine tool applications. This knowledge enables a motor manufacturer to build a more economical motor of smaller size, and consequently reduce the expense of applying motors to machine tools.

When considering the application of a motor the respective merits of the constant-speed motor and the variable-speed motor require careful analysing. With a constant-speed motor it is, of course, obvious that speed variation must be obtained by mechanical means, such as the use of a gear box. A motor running at a constant speed will develop a constant horse-power and need only be a comparatively small one, and as only a simple starting switch is necessary, the cost of the electrical apparatus is at the lowest. With the variable-speed motor, however, very few mechanical changes are necessary; thus the gearing is of the simplest and therefore cheapest construction, but this is offset by the motor and electrical apparatus needed. As the motor runs at varying speeds, it must of necessity be larger in order to develop the full power at a slow speed, consequently the motor costs more, and also the necessary controlling switch-gear. The two types of drive are, therefore, fairly evenly balanced as far as first cost is considered. The great advantage of the variable-speed motor, however, is its great range of speed adjustment in small increments, so that practically the exact cutting speed can be obtained for any operation with its corresponding advantages.

Variable-speed motors are built with speed ranges

of 2 to 1, 3 to 1, and in some instances 4 to 1. A simple illustration shows how few mechanical changes are required over a wide range. Assuming the motor to have a variation of 3 to 1, and the highest spindle speed to be 99 R.P.M., we should get down to 33 by the use of the controller alone, then

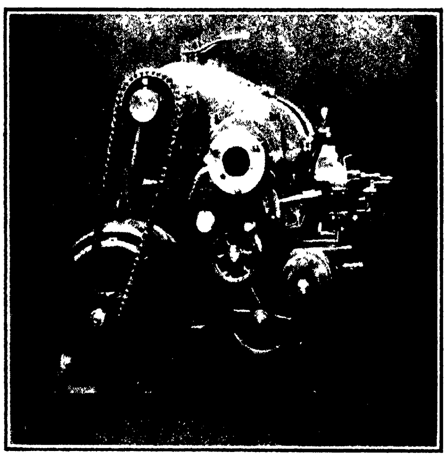


FIG. 65.—Electrically-driven head. Motor on floor.

by a change of gear we can go down from 33 to 11 by the controller, and then by another change of gear from 11 to 3·66.

Mounting of Motors.—Wherever possible it is desirable to mount the motor on some part of the machine. Here the all-gear head shows to advantage, because in many designs the top part of the

cover can be provided with machined faces to which a motor can be attached at any time. If not convenient to mount the motor on the machine it should be carried on a base-plate bolted to the machine, as shown in Fig. 65, which shows a constant-speed motor driving an all-gear head by Messrs. G. Swift & Sons, Ltd., Halifax; and in cases where this is

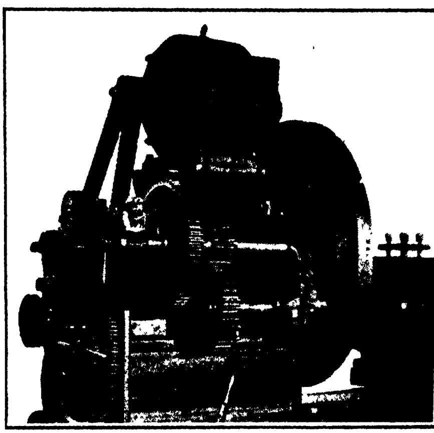


FIG. 66.—Electrically-driven head. Motor on head.

not practicable, and the motor is intended to drive some part of the machine direct, a flexible coupling should be used between the motor and machine in order to nullify the effects of any out-of-alignment of the parts.

A typical example of a large lathe head with motor mounted above is shown in Fig. 66. The headstock

is an example of treble gear, and one in which the gearing has been retained and the cone pulley substituted by a variable-speed motor. This is of 25 horse-power, and has a speed range of 1250 to 500 R.P.M.

In these two examples **a chain drive** is used to transmit the motion from the motor to the first motion shaft of the machine. The silent chain is very popular for motor drive on account of its flexibility, because its use allows the motor to be placed in any desired position, and where it is not always feasible to connect direct by gears owing to the centre distances being too great. Chain driving being positive, tension is not required to make chains grip their sprockets, as is the case when belts and pulleys are used. There is with them therefore an entire absence of slip, the minimum of journal friction, and a more economical use of power. Silent chains are run regularly up to a speed of 1250 ft. per min. Ratios as high as 1 to 6 may be used, and although a 15-tooth sprocket may be used, higher numbers of teeth are preferable as they are less severe on the chain, and help to give a quieter drive. An odd or "hunting tooth" in the pinion is also an advantage. An inclined or horizontal position with the *tight side of the chain on top* is the best, and where possible the larger sprocket should be uppermost, so that the weight of the chain is distributed over a good number of teeth.

When the motor shaft is geared direct, a gear made of some non-metallic substance should be used so as to enable smooth and quiet running. It is not possible to get entire noiselessness with two metal

gears working together, however accurately these may be cut. Materials available for use are raw hide, compressed paper, and fabroil. As a general rule, in a set of two gears it is only necessary to

FIG. 67.—Motor-driven work head on Churchill grinding machine.

have one of non-metal, in which case the mating gear may be of the usual materials. The strength and capacity for transmitting power of such gears are at least equal, if not superior, to those of cast-iron gears. Where an intermediate is used to obtain

quiet running, the intermediate should be of non-metal and not the motor pinion, because this is in mesh with both driving and driven gears, whereas the driving pinion would only be in mesh with one gear; and the periphery speed of the teeth of the intermediate and driven gears being the same, there would be as much noise as if no non-metallic gear were used.

Fabroil Gears have proved a most efficient means of drive. They are made of a cotton fabric, compressed under hydraulic pressure, and held in compression by steel "shrouds" or side-plates, threaded rivets passing entirely through both shrouds and fabric. After the blanks are machined they are impregnated with oil, rendering them when in use self-lubricating, impervious to moisture, and proof against atmospheric changes; so that apart from noiseless running they are undamaged by oil or water, and the material being elastic absorbs shock and causes the meshing teeth to come into perfect contact across the whole face. This also results in good wearing qualities, as the teeth, after running for a short period, assume a highly polished surface, reducing friction to a minimum, and giving the gear an exceptionally long life.

An interesting motor application is shown in Fig. 67. This is for driving the work head of a **grinding machine** made by the Churchill Machine Tool Co., Ltd., Manchester. It is a 15 horse-power variable-speed motor with a 3 to 1 variation. The speed controller for this motor is situated at the front of the machine within easy reach of the operator. Owing to the table and motor traversing along the

bed, it has been necessary to provide a flexible connection to the current supply. The cable used is

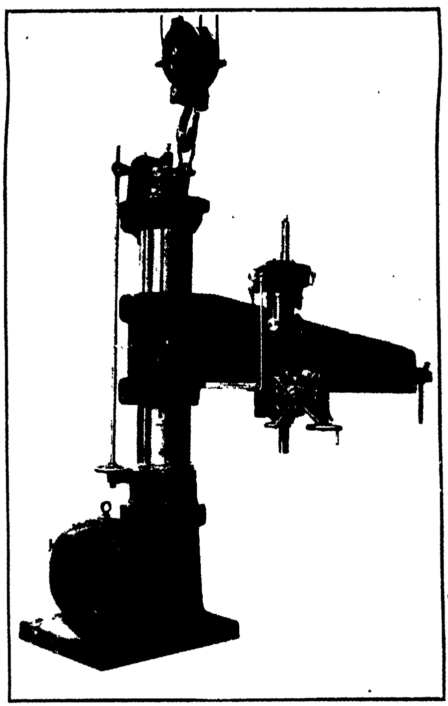


FIG. 68.—Asquith portable radial drill.

unusually flexible and is encased in a rubber sheath. As the motor moves backwards and forwards the

cable coils on or off a spiral-grooved drum, and is kept moderately taut by a suspended weight in the tubular support carrying the drum. The drum runs on ball journal bearings and is fitted inside with collector rings to make the connection between the rotating and stationary parts. *It should be noticed that the motor axis has been placed at right angles to all axes of high-speed shafts on the machine, in order to nullify any motor vibration at varying speeds.*

The Asquith **radial drill**, illustrated in Fig. 68, is of the portable type, designed for moving about the shop by the crane to avoid transporting heavy work, and therefore it was essential that the driving arrangement be as compact as possible. It will be noticed that the pillar of the machine has an extended base to which the motor is securely bolted. The armature shaft of the motor extends inside the pillar, and is connected with a vertical internal driving shaft by steel mitre gearing. The motor is of the variable-speed type, with a speed range of 900 to 300 R.P.M., and develops 12 horse-power. The spindle speeds vary from 360 to 33 R.P.M.

A very neat method used by Messrs. Asquith is shown in Fig. 69. The motor is bolted to a facing provided on the radial arm, and moves up and down the pillar with it. The horizontal driving shaft at the back of the arm is driven by bevel gears from the motor, which is of the same power and speed as in the previous example.

The drive of a **band-sawing machine** by Messrs. Noble & Lund, Ltd., Felling-on-Tyne, is shown in Fig. 70, and is an interesting example of direct drive

without any speed variation. From the motor the motion is transmitted through a spur gear reduction

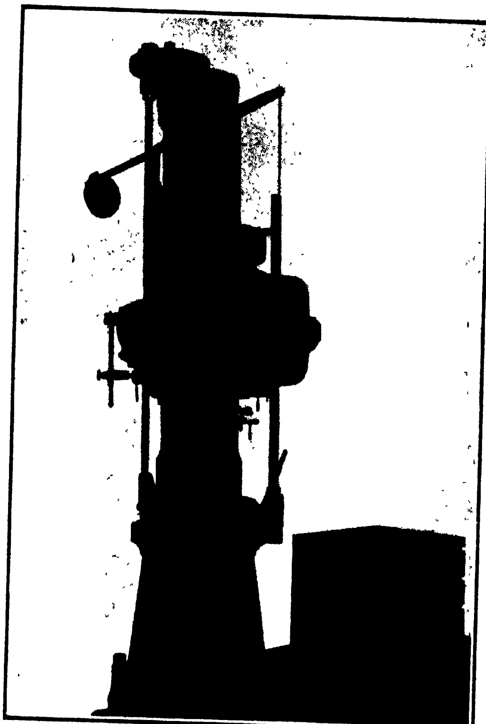


FIG. 69.—Asquith radial drill with motor mounted on arm.

to bevel gears on a short vertical shaft, then again through bevel gears to bring the drive horizontal,

and through a pinion direct to a gear attached to the band-saw wheel. The position of this drive on the top of the machine causes the driving gears to be away from contact with dirt and cuttings, which is a desirable feature. When the gears are at the lower part of the machine this contact is unavoidable.

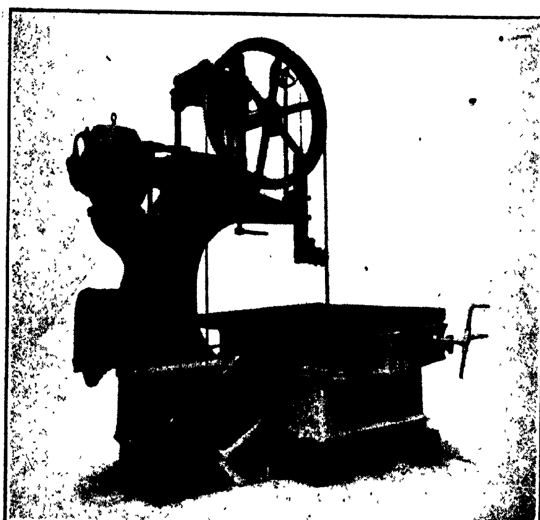


FIG. 70.—Noble & Lund's band saw.

Motor Starting Switches.—The necessary electrical equipment for a motor consists essentially of (1) A main switch of the double-pole type, which when open renders the motor circuit dead; (2) double-pole fuses connected on the "dead" side of the double-pole switch, in order that they may be replaced or adjusted with perfect safety when the

double-pole switch is opened. This point is important as it is one that is frequently neglected; (3) a motor starter switch, to allow the motor to start from rest and attain full speed gradually without undue strain on either armature or field coils.

It is advisable to have this fitted with both overload and no voltage release, in order to obviate danger

FIG. 71.—"Igranic" motor starter switch fitted with no-volt and overload release.

to the motor through either excess of current or sudden failure of electrical supply. With larger motors, where sudden overloads frequently occur, an automatic circuit breaker, set to operate before the fusing-point of the fuses is reached, is a distinct advantage. This ensures safety, whilst saving a good deal of time otherwise lost in replacing fuses unnecessarily.

Such a starting switch, as made by the Igranic Electric Co., Ltd., is shown in Fig. 71.

The neatest way of having the various parts of the switch gear is to enclose them in a panel, as shown in Fig. 72. The parts are assembled on an angle iron frame secured to a cast iron base plate. The whole being surrounded by a sheet-iron cover. Hinged doors at the front, fitted with glass windows, permit examination of the switch parts, and additional doors at the back give access to the resistances.

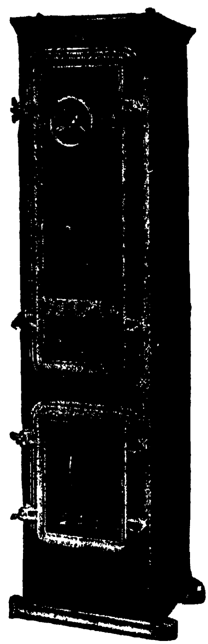


FIG. 72.—"Igranic" motor starting panel for variable speed motors.

An Igranic double-pole circuit breaker is used in this panel, designed to open the motor circuit on both poles and capable of doing this under overload conditions. It consists essentially of two single-pole magnetically operated main switches, protected by a powerful magnetic blow-out, and interlocked with the starter so that they cannot close except when all the starting resistance is in circuit. Being magnetically operated, no handle is required for the circuit breaker. The

first movement of the starter handle in the starting direction causes the circuit breaker to close, but it will *not remain closed if an overload exists* even though the operator should continue to move the starter handle forward. All circuit breaking, whether resulting from an overload or any other cause, is performed by the circuit breaker and cannot take place on the contacts of the starter, because they are "dead". Circuit is broken in a fire-proof chamber, consequently arcs are instantly quenched.

To guard against injurious overloads an "overload preventor" is provided in series with the operating coils of the circuit breaker. This consists of a solenoid whose coil carries the main current of the motor. The armature in this solenoid may be adjusted for a large variation of load. When the load for which it has been set is exceeded, the armature is lifted by the strength of the magnetic pull and opens the circuits of the coils of the circuit breakers, causing them to open both poles of motor circuit. It is an advantage to have an *ammeter* placed on the top of the panel. If the motor efficiency is known, the ammeter can be calibrated accordingly and direct readings of the actual horse-power absorbed by any particular cut can be taken.

The above apparatus is used for constant-speed motors, but when required for variable-speed motors an addition is required in the form of a shunt-field regulator. This is shown in Fig. 73, and is included in the panel shown in Fig. 72. The field regulator is interlocked with the starter so as to prevent starting the motor with resistance in the field of the motor. It is operated by the small

hand-wheel at the top of the panel, and by turning same gradually cuts out or inserts resistances in the shunt field of the motor, thereby regulating the speed of same. A great number of steps can be used so as to give very gradual increments, which, when so desired, can be arranged in either arithmetical or geometrical progression.

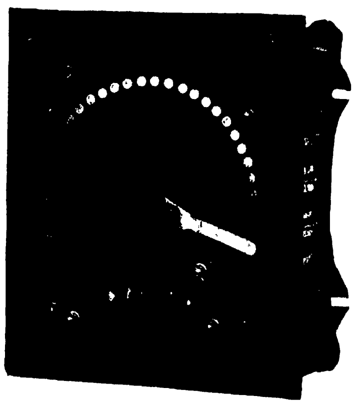


FIG. 73.—"Igranic" shunt field regulator.

Automatic Control.—This is a valuable feature on all motor drives, and in this case it is necessary to have an automatic starting switch. This automatic switch may be placed in any convenient position, either on the machine or on any convenient place away from the machine, the starting and stopping being effected by a push-button control box. Such a box can be fixed on any part of the machine convenient for the operator, or two or three of these boxes may be placed at different parts of the machine

so that it may be controlled from any position and by very simple means, for nothing can be easier than to press a button. The advantage of this will be readily seen when such a box can be fitted to the saddle of a long lathe. A very neat automatic control panel is shown in Fig. 74 with the cover removed.

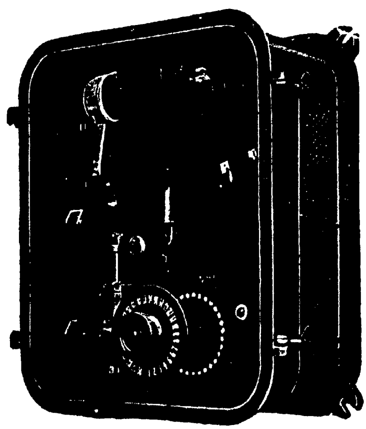


FIG. 74.—"Igranic" automatic control panel. Front cover removed.

This system has been further extended to allow the speed to be regulated—up and down—by the same means. A controller for this, also by the Igranic Co., is shown in Fig. 75, and consists essentially of a contact lever, which moves up and down over contacts. It is actuated by a solenoid and retarded by a dashpot. On the same base is mounted a magnetically operated circuit breaker of

a similar type to the one described in connection with the starting panel. The principal feature of this controller is that the contact arm does not always travel over all the contacts, but may be moved as

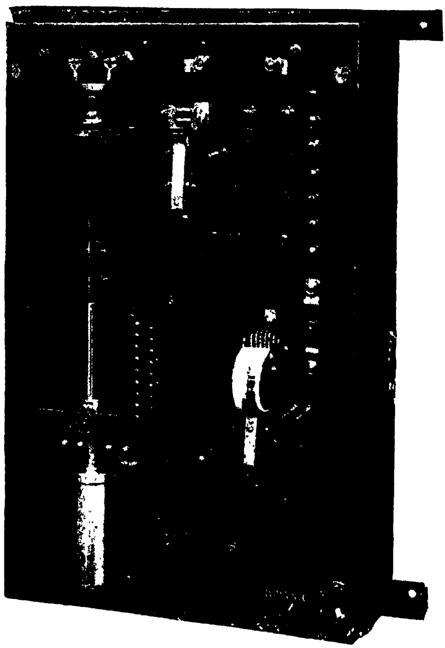


FIG. 75.—"Igranic" compensated solenoid type push button control panel.

short or as long a distance as circumstances demand, and it stays in any position for any desired period. Consequently the motor may be run for any length of time at any speed between the maximum and

minimum values, or at any instant the speed can be changed within the limits fixed by the design of the motor. This is effected merely by pressing the accelerating or retarding button.

The master control box is shown in Fig. 76. It is a small iron push-button box having four buttons marked "Inch," "Start and Accelerate," "Retard," and "Stop". Any number of these boxes can be fitted in convenient positions round a machine. To start the motor the operator presses the button marked "Start and Accelerate". If the button is only pressed for a moment the motor will run at its lowest speed until stopped by pressing the button marked "Stop". But if the first-mentioned button is depressed for a longer period the machine will not only start, but will accelerate until the finger is removed or until the maximum speed is attained. If the finger is removed before the maximum speed is reached the acceleration will immediately cease, and the motor will continue to run at the same speed as that attained at the particular moment when the finger was removed, and it will not change until either the "Accelerate" or "Retard" button is depressed. On depressing the button marked "Retard" the speed diminishes gradually until the finger is removed or until the lowest speed is reached. The motor can be stopped instantly by pressing the "Stop" button. "Inching" by means of this con-

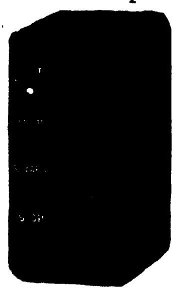


FIG. 76.—"Igranic" push button control box.

troller is very easy. It is only necessary to press the button marked "Inch". This causes the motor to run at its lowest speed so long as the button is pressed. The system is fool-proof, for the operator cannot cut out resistance too quickly. The stopping button is always a master button, that is to say, if any other button should be pressed while the "Stop" button is pressed, the motor circuit will be opened. Whenever the motor is stopped with the "Stop" button or by the operation of the overload release the contact lever returns automatically to the place where it inserts all resistance in the circuit ready for the next start, and the motor circuit cannot be closed again until this has happened.

The advantages of such control are obvious. By its use a careful operator can save his cutting tool, reduce stresses on his machine, and save current. At a hard place he can slow down, and speed up again immediately the hard place is past. Also he can save time by speeding up between cuts, if any such intervals occur in the cycle of operations. This point is referred to again in the following chapter.

Electrical Horse-Power.—In measuring the power used for motor driving, a unit is used termed a *watt*. It is equivalent to the rate of doing work when a current of 1 *ampere* flows between two points having a difference of pressure of 1 *volt*. For example, if the current flowing through a motor was 60 amperes at 220 volts, the number of watts would equal

$$60 \times 220 = 13{,}200 \text{ watts.}$$

The relation between horse-power and the watt is

$$746 \text{ watts} = 1 \text{ horse-power},$$

so that in the above example we should have

$$\frac{13{,}200}{746} = 17{\cdot}6 \text{ horse-power.}$$

CHAPTER VI.

PLANING MACHINES DRIVES.

A PLANING machine is, as its name implies, a machine for producing a plane surface, and in most cases has a table supported on a bed along which it has a reciprocating motion. The methods of obtaining this reciprocating motion are those with which we are now concerned. Owing to various difficulties, it has not been found generally practicable to cut on both strokes, and most planing machines are arranged to cut on one stroke and run idle upon the return stroke. When a planing machine has taken a stroke without doing any work, it unprofitably consumes time as well as power. The loss of time may be minimized by giving to the reciprocating table a very rapid return stroke; but the time so saved is saved at the expense of power, because in a reciprocating machine the power required to overcome the inertia of the moving parts, to arrest their motion, and impart fresh motion in an opposite direction is probably much greater than the power consumed in taking a cut.

The most commonly adopted method of driving a planing-machine table is by a rack, this giving a smooth, uniform motion with the smallest amount of power lost in friction. In some of the earlier

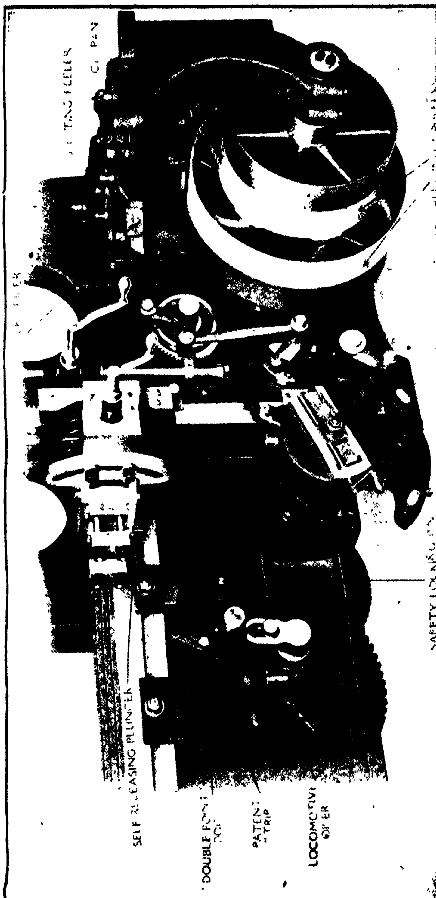


Fig. 77.—Operating side of the "Gray" planer.

machines a screw and nut were used, but with high-speed machines a considerable amount of power was found necessary to drive the screw, and it was gradually abandoned in favour of the rack.

The old-time machines were driven by a single belt with three pulleys, and at each reversal it had a long movement, passing from one outside pulley over a loose pulley to another one, thus taking considerable time to reverse the motion, and as on the return stroke one pair of gears ran idly, the arrangement was very noisy and unsatisfactory. Consequently the two-belt drive was introduced to do away with this, and is now found in all machines of medium size.

Fig. 77 shows the operating side of one of the "Gray" planers.

The use of two belts enables these to be of narrow width and to run at high speeds. They run in opposite directions and permit easy change and control of the speed for the forward-and-back motions of the table by altering the sizes of the pulleys. It also shortens the shifting range of the belts to one instead of two widths, laterally, and does not permit them to interfere and screech by both being in contact with the middle pulley at the same time.

The differential shifting gearing for the belts came in with the double belts, and was, of course, a sequence of that method of driving. By this mechanism one belt is always moved in advance of the other, and, as only one belt is handled at a time, it requires very little power to shift the belt. Fig. 78 shows a pivoted cam belt-shifter. When one bowl

is working in the cam race the other bowl is riding stationary in a circular path.

To shift a belt from one pulley to another requires that it should be deflected edgewise, so as to wind in a spiral path across the faces of the pulleys; hence the ease with which a belt can be shifted is inversely as its width and directly as the time and velocity of movement.

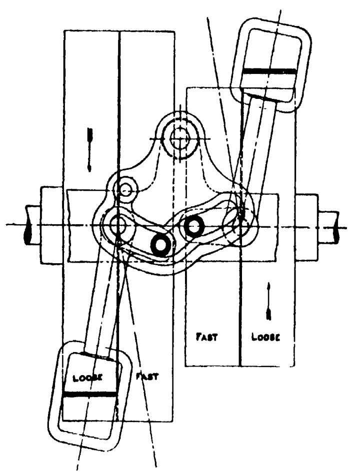


FIG. 78.—Pivoted cam belt shifter.

The belt speeds usually adopted are in the neighbourhood of 2000 ft. per min. and 4000 ft. per min. on the cutting and return strokes respectively. It should also be remembered that on the return stroke the pulley for driving the cut is running idle, and being usually larger in diameter than the return pulley its rim speed will be accelerated. With steel

rims this speed may be as high as 5500 ft. per min., but should not exceed this amount.

The speed of the table on the cutting is usually 40 ft. per min. and on the return stroke varies with the size of machine, as follows:—

3 and 4 ft. wide . . .	100 ft. per min.
6 ft. wide	93 "
7 ft. wide and upwards . .	80 "

Excessively high speeds of return are not recommended for belt-driven machines, as they are not as economical as would be supposed.

A good example of the gearing of a planing machine is shown in Fig. 79, and is from the practice of the Cincinnati Planer Co. All the pinions of the driving-train are keyed to the gears so that the shafts are relieved from all torsional strain. The pinions are of steel and the gears of "semi-steel". The bearings consist of large solid bushings, accurately ground and fitted in holes bored in the bed of the machine. By this means good bearing surfaces are ensured for the shafts, free from blow holes which are usually present in the holes in large castings.

The Bull Wheel.—It will be noticed that use is made of the bull wheel, or large intermediate gear, in the train of gearing. This meshes directly with the rack underneath the table. Its adoption in a planing-machine drive—or other machines having a reciprocating table—has given rise to much discussion. In the old days, when planing machines were first introduced, the logical method of construction was to use a rack pinion meshing directly with the rack. It was the simplest method, because

of the smaller number of detail parts required, and so great were the benefits obtained from the use of planing machines that no attempts were made towards any different construction for quite a long period.

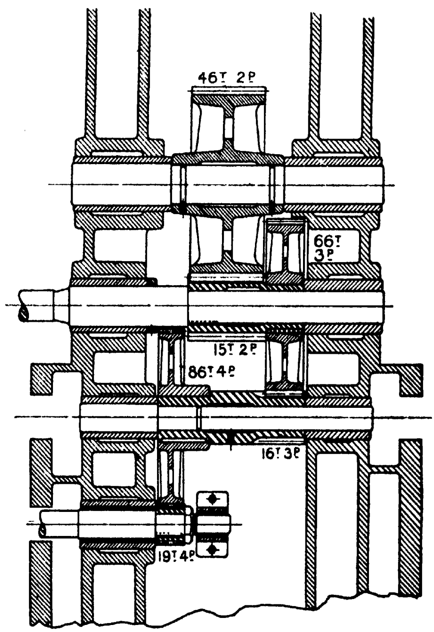


FIG. 79.—Gearing of Cincinnati planer.

It should be borne in mind that in those early days gear cutting was a luxury, and prejudice was so great that machine-cut teeth were considered inferior to cast teeth, and this, even among machine

tool makers. Also, that the involute form of tooth had not come into general use, the cycloidal tooth being used most extensively. When engineers began to make greater demands on the makers of planing machines to obtain a higher quality of work and at an increased rate of speed, it was sought to obtain an improvement in the driving arrangement. It was found that the rack pinions with the cycloidal tooth were not always satisfactory. The pinions were made of coarse pitch with a small number of teeth, and the trouble experienced, particularly on the smaller machines, was minute irregularities or wave marks on the surface of the work. This was due to the action of the teeth of the pinion, the pressure from which—due to the form of tooth—varied as each pair of teeth came into engagement from a vertical upward component to one wholly horizontal, or nearly so. The vertical component at first contact with each pair of teeth tended to relieve a part of the bearing pressure on the V's, and had the effect of lifting the table to a minute degree.

To overcome this defect pinions were used with stepped teeth, on the reasoning that by doing so, irregularities of the rack motion would be reduced, and a smoother motion to the table be obtained. The stepped tooth, of course, has its equivalent in a modern machine-cut gear with a fine pitch of tooth, which thereby has a greater number of teeth in mesh at one time than a wheel of corresponding diameter has with a coarser pitch of tooth. Thus, quite naturally, the large intermediate wheel to mesh with the rack followed as a consequence of the above reasoning, and with the belief that, on account

of its size and arc of contact with the rack, it would exert a pressure on the rack teeth that is almost constant and horizontal, and, therefore, with no lifting tendency.

But with the universal use of the involute form of tooth this reasoning is changed. According to the theory of involute gear-tooth contact, actual contact between the teeth of mating gears can only take place along the line of action, i.e. the line of pressure, and $14\frac{1}{2}°$ usually with the centre point of contact (not with the centre of each tooth), and between the limiting points, which are the points of intersection

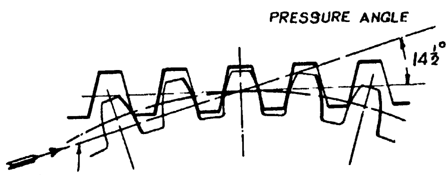


FIG. 80.—Showing contact of bull wheel tooth.

of the addendum lines and the line of action (see Fig. 80). The maximum length of contact then, for externally meshing gears, would be when the radius is infinity. Consequently, with either the bull wheel gearing into the rack, or the pinion gearing into the rack, the pressure angle is the same, quite independent of length of arc in contact.

Now, it has already been conceded that the smoothest action is with the greatest number of teeth in mesh; therefore if we had two small pinions meshing together we should not expect to get as smooth running as with two large wheels, and to carry this reasoning to infinity, we should have a

rack meshing with a rack to give the perfect result. On a planing machine drive there must be a pinion for the final motion, whether direct or through the bull wheel.

Following this reasoning out, then, if a pinion is meshing with a wheel, there cannot possibly be as long an arc of contact between them as there is between the same pinion and a rack, and, therefore, the smoother motion would result from the pinion in mesh with the rack. Chatter marks are found on many machines fitted with the bull wheel.

Extended experience has shown that the bull wheel is not at all a necessity as it may have been in the old days, but it has its usefulness for constructional purposes. For many smaller sized machines there is no necessity for it. The use of the rack pinion alone when accurately cut, and having a good outline of tooth (the number of teeth should be as many more than twenty as practicable), meshing correctly with the rack is justified, and proved by the fact that most of the medium and smaller-sized grinding machines are so arranged. This type of machine has a comparatively light table, and any lifting tendency on same would be fatal to accurate work; but, as is well known, a grinding-machine table is probably one of the smoothest-running pieces of machine tool mechanism.

For larger machines, however, it is often an advantage in laying out a train of gearing to be able to make use of the bull wheel. First it permits all the driving gears being placed inside the bed, where they can be rigidly supported by two bearings, one on either side, which is very desirable, especially

for high-speed and heavy planing. Secondly, this eliminates all overhung gearing, which construction would have to be used if the pinion were made to mesh directly with the rack.

Summarizing the foregoing remarks, it will be quite evident that the governing factor in design whether the bull wheel shall be used or not depends on constructional details alone. The whole success of a smooth-running table depends neither on the bull wheel nor the rack pinion, but on the gearing as

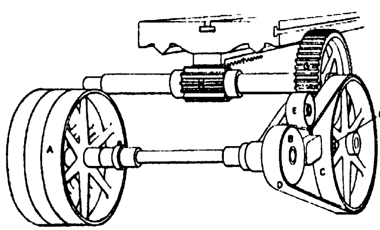


FIG. 81.—The Whitcomb second-belt drive.

a whole, which must be of good tooth outline, accurately sized and meshed, and rigidly housed.

An instance of the rack pinion meshing directly with the rack is in a small machine known as the "Whitcomb second-belt drive". Believing that a smoother motion would be given without gearing, this firm use only one gear reduction, the quick-running gears being dispensed with and a belt substituted. This "second" belt is so proportioned as to be capable of transmitting easily all of the power delivered by the driving-belts, and is, therefore, practically positive. It is automatically kept at proper tension by a weighted idler pulley. The dia-

gram, Fig. 81, shows the principal features. A indicates the usual set of driving pulleys driven by a pair of narrow shifting belts in the ordinary manner. D is the "second" belt running upon driving pulley B and driven pulley C. E is the idler pulley which follows up the belt as it stretches and keeps it at proper tension. F is a pinion attached to pulley C, and G a driven gear mounted upon the rack pinion shaft.

Variable Cutting Speeds.—A planer may be called upon to work on different kinds of material, of all hardnesses, and with tools of all qualities. All sizes of cuts may have to be taken; all kinds of finished surfaces may have to be produced. The work itself may be of all grades of stiffness, and of any weight. All these different conditions call for a planer which should be flexible. It should be possible to plane slow or fast, and do this without spending much time in changing the drive.

In many makes a gear box is carried above the machine, and thereby forms a variable-speed countershaft. Here we find again a difference of opinion, and the tendency to maintain the use of belts as giving a smoother and steadier drive than gears. This is exemplified in the Gray four-speed countershaft, Fig. 82, in which two shafts are used, driven by an endless belt running on four-step cones. A special shifting device enables the belt to be thrown from one step to another by the simple pulling of cords. The first-motion shaft runs at a uniform speed and carries a pulley to drive the quick return motion.

To change the speed, cord A is pulled which raises

the swinging jockey or tension pulley and thereby slackens the belt; then either B or C are pulled,

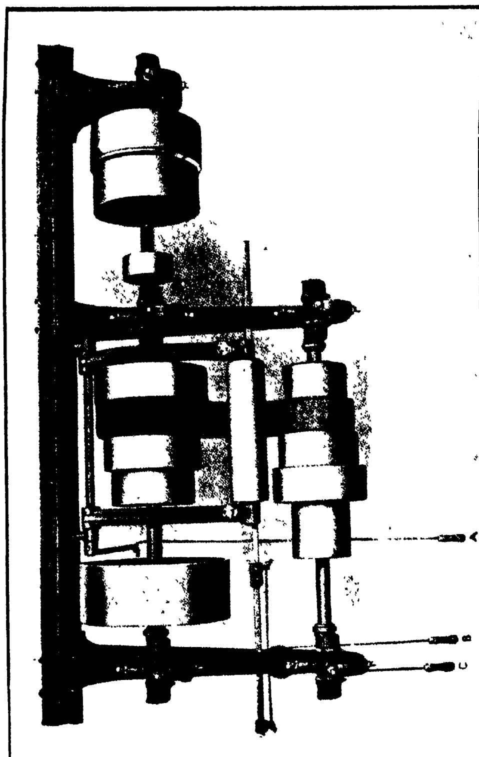


FIG. 82.—Four-speed counter-shaft for a planer.

depending on whether the speed is to be increased or decreased.

The Sellers' Motion.—The spiral-geared drive, the original patent of Wm. Sellers in 1862, is extensively used for planing-machine tables or other slides having a reciprocating motion. The rack is actuated by a short section of screw, see Figs. 83 and 84 by the Gray Co., placed upon a shaft at such an

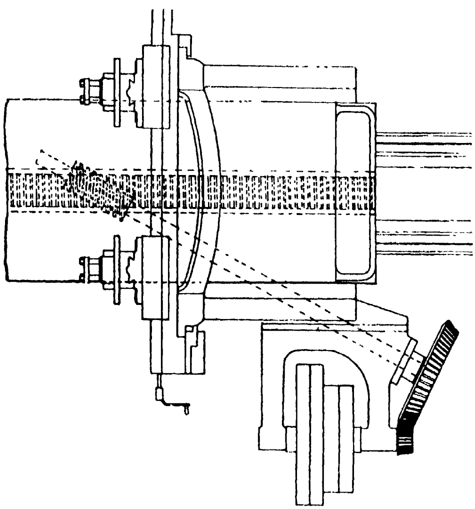


FIG. 83.—Spiral-pinion drive.

angle that the teeth on the table rack would properly engage with the threads of the screw, or spiral pinion. As the spiral pinion has at all times in the Gray Co.'s practice *eight* teeth in working contact, the pressure on any one tooth is correspondingly reduced. The continuous semi-rolling, semi-sliding action gives a drive that for smoothness of motion and freedom of

vibration is unsurpassed. It is due to this fact that there is practically no wear either of the rack or pinion even after years of use. The end thrust of

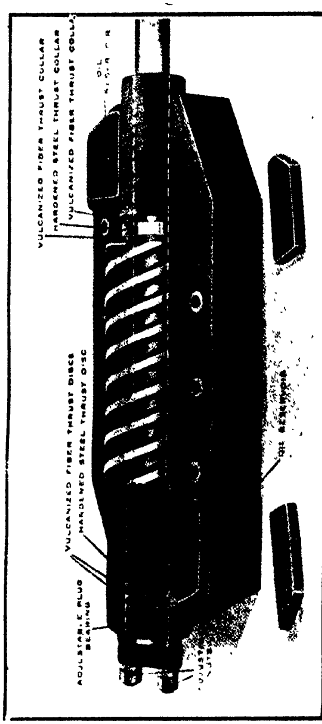


FIG. 84.—The Gray spiral pinion.

the spiral pinion is taken by alternate hardened steel and vulcanized fibre collars, and wear of these is taken up by an adjustable plug bearing. Two large

oil reservoirs are provided, so that the whole box and bearings are flooded with oil.

Disadvantages of Shifting Belt.—There are, of course, as with all other mechanisms, certain disadvantages of the shifting-belt mechanism. One is, that for heavy machines, the belts, when made very wide, require considerable force to move them, and being slower in moving across the pulleys, there is time lost in reversing. This difficulty is partially overcome by making a double set of driving pulleys and dividing the width of the belts between them, as shown in Fig. 85.

Perhaps a more serious objection is, that for high speeds of return there is so much momentum to be absorbed and then re-initiated at each reversal. The troublesome momentum which has to be overcome is not in the table, but in the stored energy in the rapidly running pulleys and gears, and the difficulty is not so much the stopping of the running parts at the end of the return stroke as in *starting them at the beginning of the return stroke.*

By its own momentum a table would move through a very short distance. A simple example will explain this. In a planing machine of the ordinary type with a two-belt drive and pulleys of different sizes, with light steel rims, the peripheral speeds of the rims are 4700 ft. and 3510 ft. per min. respectively for a table speed of 100 ft. per min.

Weights of 14 lb. in the larger and 10 lb. in the smaller, distributed in the rims of the pulleys, would require expenditures of 1330 and 530 ft. lb. of energy to accelerate them from a state of rest —a total of 1860 ft. lb. Obviously, the energy

required to accelerate the entire gear would be represented by a much higher figure.

The power required to accelerate a table, 3 tons in weight, to a speed of 100 ft. per min. is 290 ft. lb.,

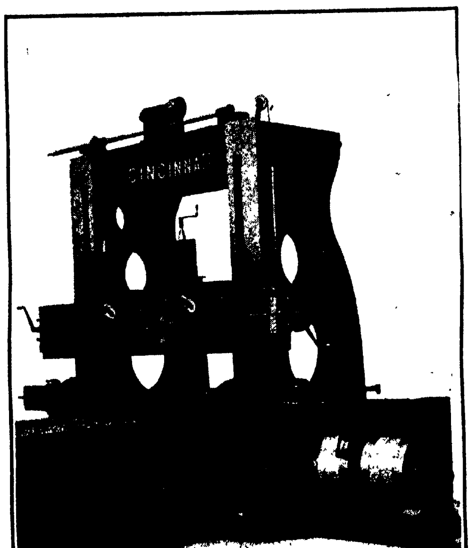


FIG. 85.—Cincinnati planer with double set of driving pulleys.

so that there is 6·4 times as much energy in the pulley rims alone as in the table.

This is an example from good practice, and we can easily see, when we consider the formula $\frac{Wv^2}{2g}$, how

this ratio would very rapidly increase with any increase in the weight of the pulley rims, and particularly in the peripheral speed, as the latter increases as the square of the velocity. With heavy cast iron pulleys cases are known where the stored energy of the pulleys is as much as 40 to 50 times higher than that of the table.

Even taking all this into account, a belt seems to be a most efficient medium in dealing with this problem. It comes on to the pulley gradually (comparatively speaking), slipping a little at first, and slipping less and less until the speed of the pulley equals the speed of the belt. The operation of the belt is interesting, as well as peculiar. As its slip increases and the belt and pulley get warm, the hug of the belt is closer. In short, a belt has all the desirable features for starting and stopping a load without shock, and it is, and will continue to be, incorporated in many of the best designs.

To reduce as far as possible the fly-wheel effect of the machine pulleys, the rims should be made of steel and of the lightest possible construction compatible with safety. Some of the American makers use driving pulleys of aluminium alloy to reduce the weight.

Tests made by the Cincinnati Planer Co. with 35 ft. cut and 85 ft. return with cast iron pulleys gave 165 strokes in 30 mins., while with aluminium pulleys 189 strokes were made in the same time. The stroke was 4 ft. in each case. This saving in time is due to the quicker response of the light pulleys in reversing which also makes it possible to plane much closer to a shoulder should this be necessary.

In another test with a 72-in. motor-driven planer, cast iron pulleys took 39 horse-power at the reverse while only 30 horse-power was required with the aluminium pulleys. These figures show a saving of nearly 25 per cent in power and again of 15 per cent in the number of strokes per hour.

It must be borne in mind that the table and all its gearing have to be *started from rest* at each stroke. For this reason it is an advantage to have the return pulley large in diameter, so as to have as long a leverage as possible to help in starting the parts on the quick return, and some makers use a return pulley as large in diameter as the cutting pulley. But with this arrangement there is the objection that there is an increase in the momentum which has to be overcome, owing to the increased rim speed, and also that the belt speed becomes too high for practical use. The return belt, owing to its higher speed, always exerts a greater pulling force than the cutting belt; therefore, it seems advisable to have the return pulley smaller than the cutting, and thus strike a happy medium between the conflicting factors—time of reversal and waste of energy.

It has also been found that fly-wheels on the counter-shafts give beneficial results. These are running continually in one direction, and the action of the fly-wheels is to steady the running and assist the action of the belts at the point of reversal. It is now customary to make the two driving pulleys on the counter-shaft with heavy rims for this purpose. Primarily fly-wheels were used with electric motors to preserve these from excessive shock, but their use has become extended to ordinary belt-driven machines.

The Bateman Fly-wheel Drive.—To enable a very quick retardation and acceleration of the parts at the moment of reversal, the use of a fly-wheel combined with a friction clutch, by means of which the stored-up energy in the fly-wheel is given out, is

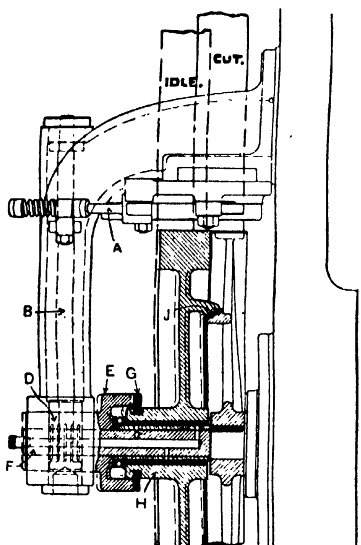


FIG. 86.—The Bateman fly-wheel drive.

the principal feature in the "Bateman" machine. This is of the shifting-belt type with the belts at each side of the machine. Each belt is wider than the face of the fast pulley, and a portion is always in contact with the loose pulley, which is in the form of a fly-wheel, and is thus kept continually running in one direction.

The arrangement is shown in Fig. 86. The fly-wheel carries a conical clutch the mating part of which is carried by the fast pulley. The fly-wheel pulley is free to move sideways along its shaft, and the motion of the belt-shifting gear is transmitted to it and moves it so that the friction clutch J is brought either into or out of action. This motion is so timed as to take place when the belt is thrown on or off the fast pulley, and, together with the overlapping of the belt from the fast on to the fly-wheel pulley, enables the conserved energy of the fly-wheel to be transmitted into the fast pulley at the moment when a large amount of power is most required.

The Mitchell Drive is a belt drive in great favour for heavy machines, and differs from the shifting-belt drive in that any width of belt may be used and any amount of power delivered to the machine. It is made as a complete piece of mechanism in itself, as shown in Fig. 87, and may be applied to either new or old planers, or to other machines requiring a reversing mechanism.

The drive is very simple, and is, briefly, a method of running two loosely-hung belts in opposite directions. Each belt is run at whatever speed is desired, and is alternately tightened by means of a jockey pulley. It has already been pointed out that there is a difficulty in shifting a wide and heavy belt sideways; and that there is a momentary loss of power at the beginning of the side movement. In the Mitchell drive this difficulty does not exist, as the two belts run in fixed paths.

The jockey pulleys operate on the slack side of the belts and cause such a large arc of contact to be

embraced by the belts on the pulleys that there is no appreciable slip between the belt and the pulley, and the machine is reversed instantly. A little slip is necessary, however, to enable a planer table to be reversed without shock, and this is provided for by means of an adjustable slipping clutch.

In Fig. 87 A is the first motion shaft, running at a speed of 600 revs. per min., and carrying two heavy fly-wheels B B, the energy stored up in which is given out at the moment of reversal and tends to smooth out the peaks generally seen on power diagrams. The drive is arranged for a constant-return speed and two rates of cutting speed obtained by means of sliding gears at C. The jockey pulleys D are pulled in and out by means of a rocking shaft E. At F F are ball-bearing eccentrics coupled to the swinging jockey pulley frames G by adjustable connecting rods. The rocking shaft E is operated by the dogs on the planer table. Several methods of operating this shaft by the dogs are used according to conditions. In smaller sizes the action is direct. In the larger sizes, however, the shaft E is operated by an auxiliary power, using preferably compressed air. The dotted lines at H show the location of the air cylinder, the piston being provided with rack teeth which mesh with a pinion J on the end of shaft E.

The slipping clutch is shown at K connected to the driving pulley for the cut. This form of clutch, consisting of wood blocks of a double V-section between two plates held together by spiral springs, has been found very satisfactory.

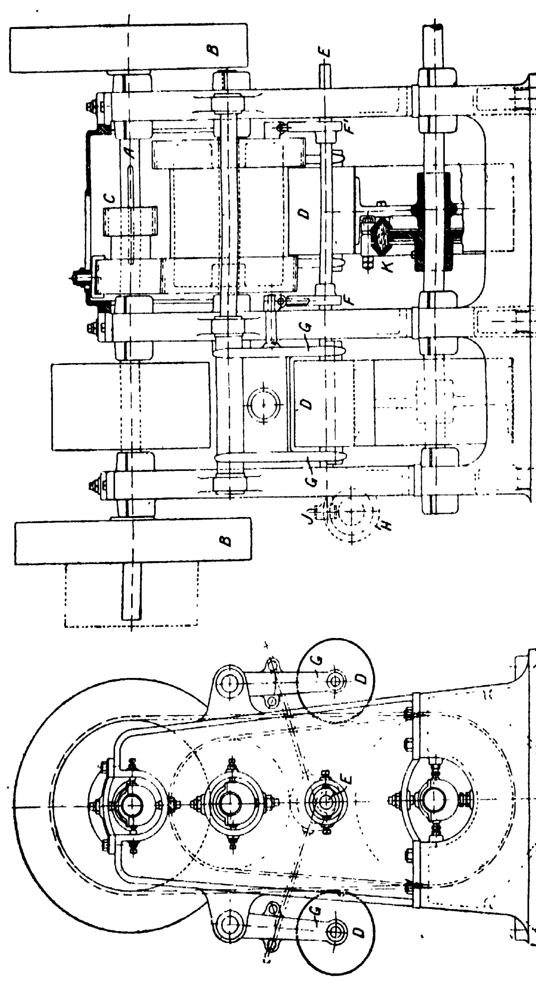


FIG. 87.—The Mitchell drive.

CLUTCH DRIVES.

Mechanically-operated Clutch.—A radical step was made by Messrs. Wm. Sellers & Co., of Philadelphia, who in 1893 patented an arrangement of drive in which only one belt is employed. Fig. 88 shows the arrangement of the clutches and controlling mechanism. The belt runs at a constant speed for returning, and drives the forward motion clutch through a suitable train of reducing gears. It should be noted that there is a pair of clutches for each motion to ensure a good drive. The clutches marked F are fixed to the driving shaft; the other clutches are all free to slide endwise, K being a double clutch and secured to the shaft by a cotter. For instance, if clutch K is moved to the right into gear M, this latter is forced against the fixed clutch F^1, and M drives the shaft through K and F^1.

It should be particularly noticed that the principal high-speed rotating parts revolve continuously in the same direction, and that there is no reversal of high-speed pulleys or parts with high peripheral speeds. The fly-wheel action of the parts whose motion is reversed, owing to their relatively small size, is then reduced to the minimum, and loss of power incidental to their reversal is avoided. This approaches very nearly to an ideal driving gear.

The clutches are conical and lined with blocks of hard wood. The sliding clutch is operated by an internal rod actuated by bell-crank levers of the familiar type found on the spindles of turret lathes. When the table comes to the end of its stroke, a

brake action is applied gradually through the clutches

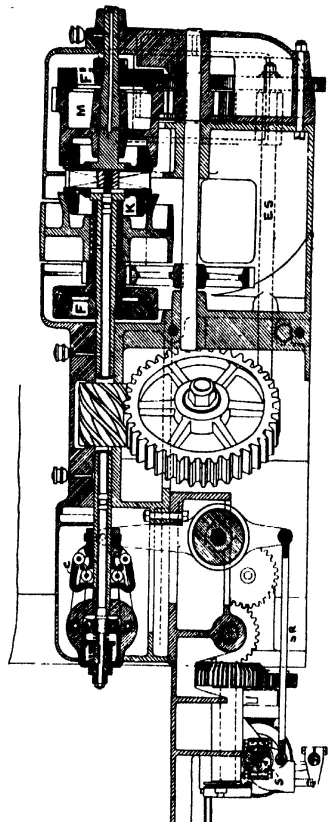


FIG. 88.—The Sellers' clutch drive.

to bring the motions to rest before the reversing power is fully applied. This acts as follows:—

When the table dog strikes the reversing lever, the shifting rod (S R in figure) is moved, and with it the lever and sliding cam C, carrying the ends of the bell-crank levers. This, acting on the internal rod, withdraws the clutches from the driving gears and presses them against the pulley with a pressure resulting from the compression of a spring so as to act as a brake and retard the motion without reversing it until the forward motion of the table has ceased. An independent friction clutch is used to grip the oscillating shafts and prevent any falling away of the rod S R after it has been once moved. The action of the dog also withdraws an abutment, and so releases an escapement which permits another mechanism to make a complete half-turn. This acts on a cam and transmits positively a further movement of the rod S R and sliding cam C, and so presses home the clutches for the return motion. The escapement train is positively driven continuously in one direction from a shaft E S driven from the driving gearing. The action of finally pressing home the clutch is thus positively operated.

Pneumatic Clutch.—The next departure by Messrs. Sellers was to substitute compressed air for the mechanical lever system employed in the last-mentioned construction for operating the clutches, and to interpose in the reducing train a pair of change wheels by which the ratio may be altered to produce different cutting speeds with a constant speed of return. This arrangement is shown diagrammatically in Fig. 89. The shaft P carries the spiral pinion driving the table, and at the outer end is a spur gear O. This is driven by the spiral

pinion N on the pulley shaft, K. The driving pulley A runs loose on the shaft, continuously, and in the

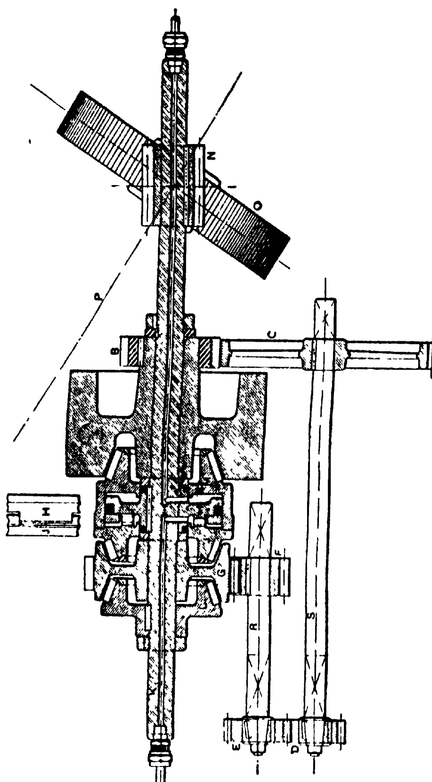


FIG. 89.—The Sellers' pneumatic clutch drive.

same direction. On the hub is a pinion B driving through a reducing train of gears C, D, E, and F to a loose gear G on the pulley shaft. It is evident

that G will run in the opposite direction to the pulley A, and at a reduced speed, depending upon the ratio of the gears in the train. The pulley is turned on one side to form the half of a conical friction clutch. G is turned on both sides in a similar manner. J and M are two conical elements bolted together and forming an air-tight cylinder free to move back and forth on the disc H, which is keyed and pinned to the shaft K and forms a piston in the cylinder. In order to compel the shaft to rotate with the friction clutch, the head of the cylinder J has notches for teeth on the surface of the piston H. These form a claw clutch, and permit end movement, while they compel the parts to rotate together. Air admitted to one end of the cylinder through the centre of the shaft K, for example between the parts H and M, will cause the cylinder to move in the direction of the pulley A, to press the friction cone against the pulley, and cause the clutch to rotate with the pulley. This movement will be transmitted through the piston H by the clutch teeth and cause the shaft to rotate in the same direction, which produces a return movement of the table. Admitting air to the opposite end of the shaft will cause J to engage with G and force the latter against the stationary clutch L, which is keyed to the shaft. G will thus drive the shaft through both the clutches. D and E are the change gears.

The compressed air is obtained either from the shop installation or from a compressor attached to the machine.

In operation, the table stops move a valve, which admits compressed air alternately to opposite ends

of the cylinder, and by regulating the velocity of the admission the speed of reverse can be nicely gauged. The table is brought to rest promptly, and started up in the opposite direction without shock. Messrs. Sellers make a complete range of these machines, from 3 ft. to 14 ft. wide. On their 8 ft. machine the belt is 11 ins. wide, and runs at a speed of 2500 ft. per min., at which speed it is capable of easily transmitting 50 horse-power.

Magnetic Clutch. — Amongst various clutches which have been tried is the magnetic clutch, made in various forms, but not so satisfactory as the pneumatic clutch. A magnetic clutch usually consists of a pair of electro-magnets and a disc which is attracted now by the one, and then by the other, according to which magnet receives the current. A simple reversing switch operated by the table dogs controls the current. They are usually fitted between the driving pulleys and driven by open and crossed belts. As the pulleys do not reverse, and the magnetic clutch parts are comparatively small in diameter, the fly-wheel effect of these is reduced to a minimum.

MOTOR DRIVING.

Electrical driving of a planing machine may be in two ways: (1) constant-direction motor, (2) reversible motor. In the first case the motor may be either constant speed or variable speed. The motor is generally mounted on the top of the machine, as Fig. 90, and drives through belts in the ordinary way. Sometimes two motors are used, one constant speed for the return and one variable

speed for the cut. The great advantage of this method is that a fine variation of speeds can be used best suited to the material to be operated upon.

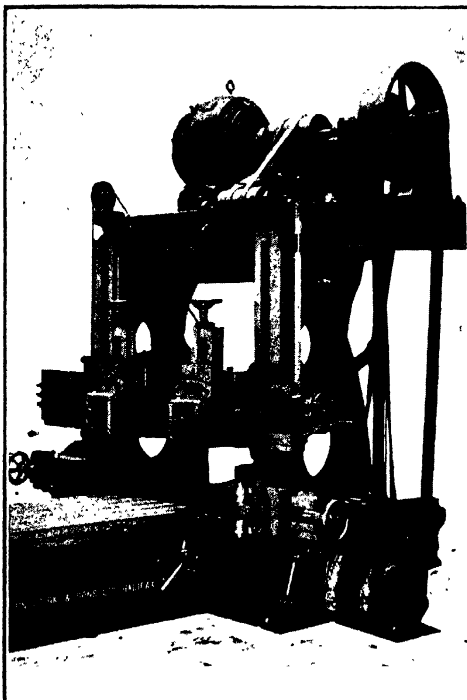
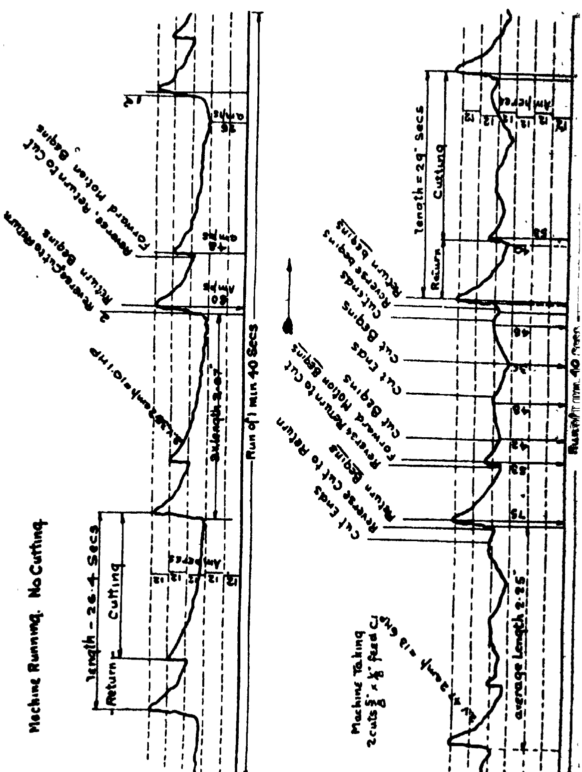


FIG. 90.—Stirk planing machine with motor drive.

Power Diagram.—Electric driving has also permitted the use of recording instruments, which show

graphically the variation in power consumption Fig. 91 shows such diagrams taken from a planer



running light and also with two tools in use. The machine was driven by a 15 horse-power compound-wound motor running at 400 revs. per min.,

with a heavy fly-wheel on the armature shaft. The results are:—

Power required on cutting stroke, no cut	10·3 h.p.
Power required on return stroke, including reversal	16·9 h.p.
Power required for heaviest cut . .	14·4 h.p.
Power required at reverse from cut to return	22·4 h.p.
Power required, average of cycle . .	13·6 h.p.
Approximate cutting speed . .	25 ft. per min.
,, return ,, . .	75 ft. ,, ,,

The power included motor and counter-shaft.

Reversible Motors.—The electrical control of a reversible motor has proved to be one of the best systems for driving a planing machine. The motor is coupled direct to the driving shaft, preferably by a flexible coupling, doing away with all belts, pulleys, clutches, and counter-shafts, and making a positive and highly efficient drive. Some of the advantages of the reversible motor are that the overload on the motor at the moment of reversal is reduced to a minimum; that it is possible to obtain very short strokes of the table without any perceptible shock, and without injury or abnormal wear to the mechanism; and that the length of stroke becomes practically positive, being accurate to $\frac{1}{8}$ inch.

A planing machine provided with a direct-motor drive and reversing mechanism of this type is shown in Fig. 92. The master switch is attached to the side of the bed and connected by a link to the shifter lever with hand control, as with the ordinary belt-driven machine; the dogs limiting the length of stroke and reversal being automatic. On the con-

troller are mounted two index dials, one to control the range of cutting speeds and the other the return speeds. The motor used generally has a speed range of 4 to 1, the revs. per min. varying from about 250 to 1000. The cutting speeds obtainable vary from 22·5 to 90 ft. per min., and the return speeds up to

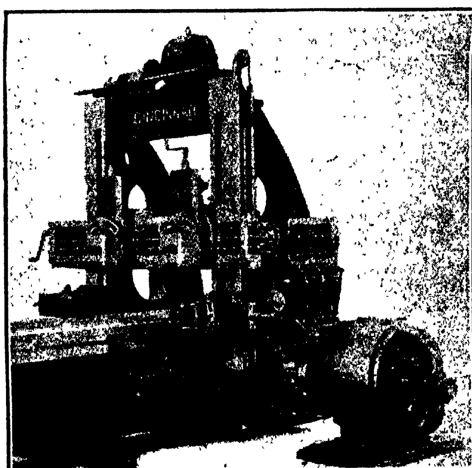


FIG. 92.—Reversible motor-driven planing machine.

100 ft. per min. The cutting and return speeds are entirely independent of each other, so that it is, for example, possible to use the slowest cutting speed and the highest return speed.

At the end of the quick return stroke, immediately before reversal, the field resistance is short circuited, which causes an infinite load to be put on the motor

for a fraction of a second, thereby bringing the armature to rest very quickly. The motor then reverses and the cycle repeats itself.

The Lancashire Drive, made by the Lancashire Dynamo & Motor Co., Ltd., Manchester, is an interesting system, the principle of which is that each driving motor shall have its own independent direct current generator to supply it with current. This generator may be driven in any convenient manner, but the usual arrangement is to have it direct coupled to a high-speed motor fed with current from any supply available, which may be either direct or alternating current.

The generator and reversible motor are both independently excited machines, and the whole series of events at the motor is accomplished by manipulating the generator field. Variation in speed is obtained by varying the field strength of the generator, thus supplying a varying voltage to the motor.

The success of the system is largely due to the fact that the generator field may be handled in ways quite impossible with the field circuit of the motor, and the outstanding advantages of the arrangement are: the simplicity of the switch gear; the wide speed range available owing to the great variation of voltage possible and practicable; and the favourable conditions under which the motor does its maximum load. These advantages, together with the fact that at the moment of reversal the table motor *actually regenerates* and returns current to the line, are held by the makers to outweigh the admitted disadvantage of using three dynamo electric machines for each planer.

There is also an **accelerating device** fixed on the side of the machine and worked by special stops on the edge of the table. On the forward strokes these stops operate the switch, but on the return stroke they glide over same. The functions of this switch are as follows :—

1. To speed up the table between cuts when two or more surfaces some distance apart are being planed. In this instance the first stop which strikes the switch speeds up the planer motor and the next one on striking the switch slows it down so as to return it to its original cutting speed before entering the next facing.

2. *For speeding-up after the tool has entered the cut, thus increasing the life of the tool.* In this case, for example, say cutting mild steel, the tool would enter at about 18 ft. per min., and when once entered would be speeded up to about 50 ft. per min., and then at the end of the stroke slowed down so as not to break the edge of the work. On considering this matter of speeding up the machine when in the cut, if we take for example the lathe, we know that higher cutting speeds can be used on this than on the planing machine. This is accounted for by the fact that in the case of the lathe, when the tool has once entered, it is on work for the completion of its travel, but in the case of a planing machine, the tool has to enter the work for every stroke, which is of course injurious to the cutting edge, and therefore a slower speed is necessary.

Referring to the difference between the two equipments the maximum range of speed with the single motor equipment is generally 4 to 1, whereas with

the three motor equipment 8 to 1 is the general rule.

For instance, taking a planing machine 5 ft. × 5 ft. × 20 ft., with the single motor, this could have cutting speeds varying from 22·5 to 90 ft. per min. and a return speed of 90 ft. per min., and with the three motor equipment cutting speeds of 22·5 to 90 ft. per min. and a return speed up to 180 ft. per min.

CHAPTER VII.

DRIVES FOR VARIOUS MACHINES.

1. **Shapers and Slotters.**—In shaping and slotting machines there are two mechanisms most generally used to give the reciprocating motion to the ram, which aim at giving an approximately uniform rate of speed for cutting and a quick speed for returning the ram on the backward stroke of the tool. These devices are the Whitworth quick-return motion and the slotted-link motion. With their use a ratio of return to cut of about 2 to 1 on full stroke and a rather less ratio on shorter strokes is the usual thing.

The Whitworth Quick-Return Motion is a well-established piece of mechanism for driving the rams of shapers. A sketch diagram of this is given in Fig. 93. Here the connecting-rod which moves the ram is marked R. It is driven by a pin F attached to the crank, and is adjustable in a T-slot in order to give varying lengths of stroke. The crank is carried by a shaft, which is eccentrically mounted in the sleeve K, this being supported by the framework of the machine. Rotating on the sleeve is another crank disc A, receiving a rotary motion through gear teeth cut on its periphery from the driving

pinion C. B is a pin with a sliding block G, which slides in the back of the crank arm.

In order to understand the reason for the alternate quick and slow strokes of the connecting-rod, it will be well to look at the diagram on the left-hand of the figure. The two circles eccentric with each other represent the paths of the two crank pins B and F, B travelling around the centre K, and F around the centre E, while at the same time G slides

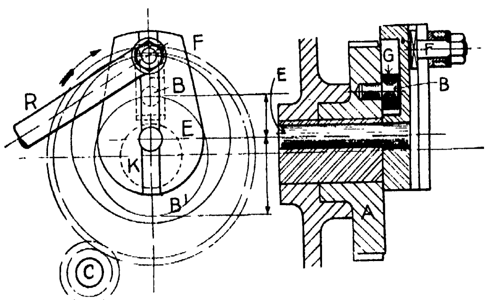


FIG. 93.—Whitworth quick return motion.

along the slot in the crank. In the position shown the crank is on the quick-return stroke.

It should be remembered that the pin B in the block G travels at a uniform rate of speed. It will therefore be obvious that at this point the speed of F will be considerably greater than that of B, because B is in operation on the shortest length of leverage, as the arm EB. On the opposite side of the circle the pin is working on a longer leverage, as arm EB′, thereby reducing the speed and also giving a more powerful effort for the cutting stroke. Any difference in the amount of eccentricity of the

two centres would, of course, result in a variation of the two speeds, and their ratio.

The Slotted-link Motion is shown in diagram form in Fig. 94. It usually consists of a crank A driving a link B which is pivoted at P. The connecting-rod between the link and the ram is represented by the line BE. The extreme positions of the link are at B*i* and B*o* when the lines are tangen-

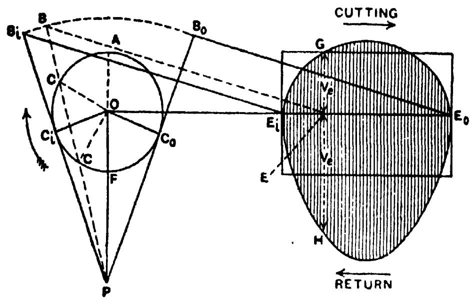


FIG. 94 —Slotted link quick-return motion.

tial to the crank pin circle as OC*i* and OC*o*. When the crank moves through the larger arc C*i*, A, C*o* the ram makes its cutting stroke, and the return is through the smaller arc C*o*, F, C*i*. As the crank pin moves with a uniform velocity, the times taken for cut and return will be proportionate to the lengths of the corresponding arcs.

In the shaded diagram the height of the ordinate at any point shows the rate of speed of the ram at that point. To obtain the speed diagram, draw the link in any position PB, and find the corresponding point E on the ram. Set out as in Fig. 95 a line

CX to a convenient scale to represent the velocity of C at right angles to the equivalent link OC. Then draw CD at right angles to PC, and XD parallel with the link. D will then be a point on the diagram.

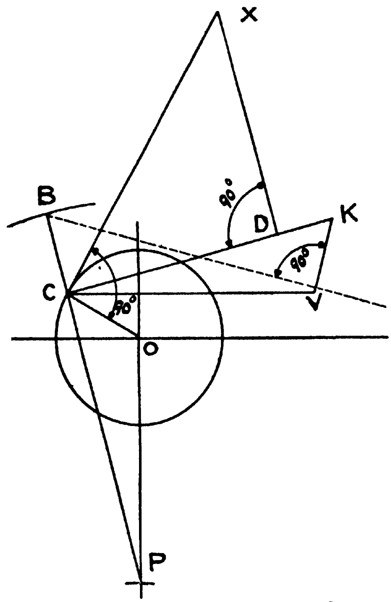


FIG. 95.—Method of obtaining velocity diagram.

Now extend CD to K making CK in proportion to PB when CD corresponds with PC. CK thus corresponds with the velocity of point B. We must now take the effect of the connecting-rod into consideration. As the ram moves in a horizontal slide, next draw a horizontal line from C to correspond

and then draw KV at right angles to the line of the connecting-rod. V will be a point on the velocity diagram, and the length CV must be measured on the scale to which CX was drawn.

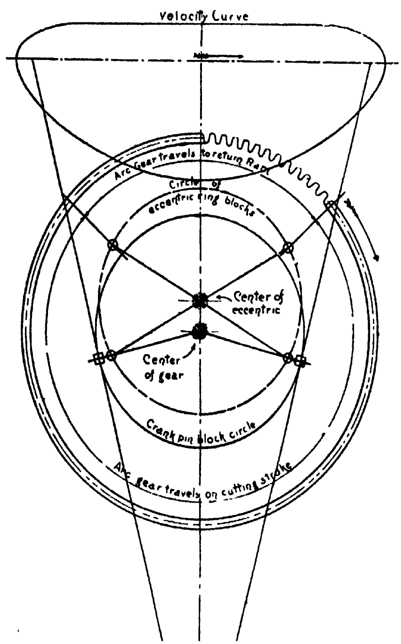


FIG. 96.—Combination quick-return motion.

Combined Motions.—It is the practice with some makers to combine the above two motions with the resultant gain of a *more uniform rate* of cutting and a considerably increased speed of return over what

is possible with either of the other two motions used alone. Such a combination is used in the pillar-type shaping machines made by the Stockbridge Machine Co., U.S.A., diagram of which is given in Fig. 96. As apparent from Fig. 94, with the regular crank and slotted-link motion, the speed of the ram

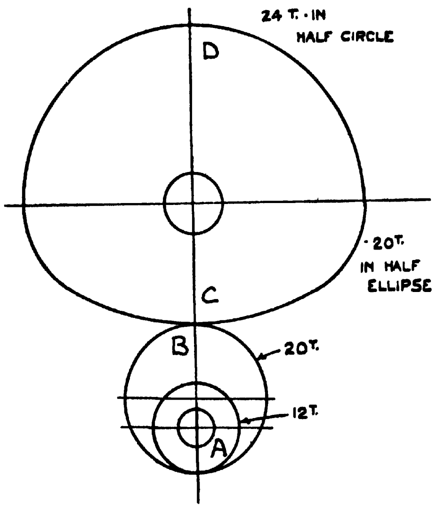


Fig. 97.—Oval-eccentric gearing.

is not uniform, but is greatest at the middle of tne cutting stroke, whereas with the combined motion it is practically uniform.

Oval-eccentric Gearing (Fig. 97).—This is a peculiar type of gearing sometimes found in crank-driven slotting machines.

On the driving-shaft there is an ordinary pinion, A

of 12 teeth, and attached to it is a pinion B of 20 teeth, but placed eccentrically on the shaft. Above these, on the crank-disc shaft is a composite gear. Meshing with the eccentric pinion is one-half of an elliptical gear C, having 20 teeth in the half, so that for one turn of pinion B the same number of teeth are exchanged, and the gear makes one-half of a revolution and gives the return stroke to the ram. At the end of the stroke the teeth of A and B blend at the same pitch-line, and the motion is then transmitted to the other half of the gear D, which is concentric and has 24 teeth in the half-circle. The engagement of the concentric pinion is thus at the same speed at which the eccentric pinion leaves it. It will be seen that as there are twice as many teeth in D as in pinion A, the latter must make two turns for one half-turn of the composite gear. This will take place at a uniform rate and be used on the cutting stroke. The ratio between cut and return is thus 2 to 1.

Boring and Turning Mills.—The boring and turning mill is a machine having a horizontal circular table which must be rotated at varying speeds. It is essentially a lathe with the face-plate horizontal instead of vertical. The conditions of driving are therefore modified from the lathe to suit the necessary constructional changes required by the machine, but the first motion and the variable speed are exactly similar to a lathe, using a cone pulley and gearing, or a gear box as desired, being placed either at the side of the machine, as Fig. 98, or at the rear. Fig. 99 shows the diagram of a 4-ft. boring and turning mill as made by Messrs. Geo. Richards & Co., Ltd., Broadheath, and is representative of a cone-driven

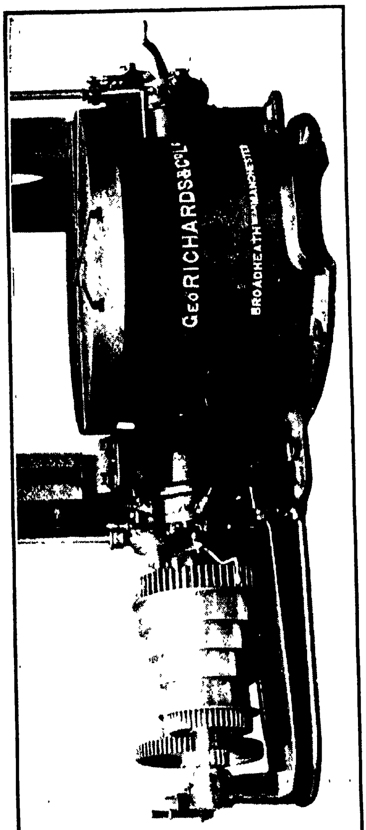


FIG. 98.—Cone pulley drive on boring and turning mill.

machine. The cone and gear are carried in a head-stock attached to the side of the machine and is of the double back-gear type, giving with a two-speed

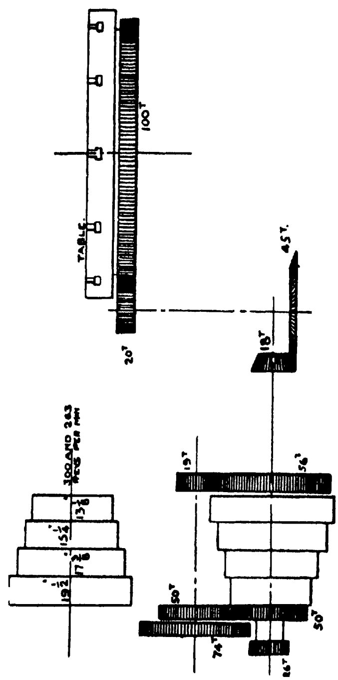


FIG. 99.—Diagram of drive of Richard's boring and turning mill.

counter-shaft twenty-four changes of speed to the table. A pair of bevel gears are necessary to transfer the horizontal motion to a vertical, after which a pinion meshes direct with a large gear bolted directly underneath the table.

As the diameters of the cone are $19\frac{1}{2}$ ins., $17\frac{3}{8}$ ins., $15\frac{1}{4}$ ins., $13\frac{1}{8}$ ins., and the counter-shaft speeds 300 and 263 revs. per min., the highest speed of the table is thus

$$\frac{300}{1} \times \frac{19\frac{1}{2}}{13\frac{1}{8}} \times \frac{1}{12\cdot5} = 35\cdot6,$$

and the lowest speed is

$$\frac{263}{1} \times \frac{13\frac{1}{8}}{19\frac{1}{2}} \times \frac{1}{8\cdot4} \times \frac{1}{12\cdot5} = 1\cdot7.$$

The full range of speeds is as follows:—

Single gear . .	35·6	31·2	27·4	24·0	21·0	18·2	16·1	14·1
Intermediate gear	12·1	10·6	9·3	8·1	7·1	6·2	5·5	4·8
Full gear . .	4·2	3·7	3·2	2·8	2·5	2·2	1·9	1·7

The ratio between highest and lowest speeds is

$$\frac{35\cdot6}{1\cdot7} = 20\,9 \text{ to } 1.$$

The maximum velocity of the driving belt is

$$\frac{19\frac{1}{2} \times \pi \times 300}{12} = 1531 \text{ ft. per min.},$$

and the minimum velocity

$$\frac{13\frac{1}{8} \times \pi \times 263}{12} = 903\cdot5 \text{ ft. per min.}$$

Worm driving is also used for boring and turning mill tables. Fig. 100 shows in diagram form a drive used by Messrs. John Stirk & Sons, Ltd., Halifax. A gear box is used, placed at the rear of the machine.

From the box the motion is transmitted through a silent chain to a sleeve running freely on shaft A. Gear B is keyed to this shaft, and connection of the two is made by means of sliding claw clutch. This gives the single gear. For the double gear the

clutch is disengaged, and the motion is then taken through gears C, D, E, and B to shaft A, and thence through another pair of gears at F to the worm shaft. A ball-thrust bearing is used to take the thrust of the worm, the worm being of the multiple-

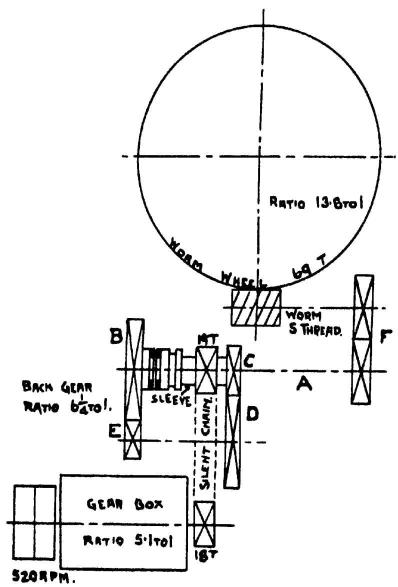


FIG. 100.—Diagram of worm-drive for Stirk boring and turning mill.

thread type designed to give a high efficiency, and with its gear runs in a bath of oil, a necessity with all high-speed worm gearing.

Drilling, Boring, and Milling Machines.—These

machines have all features in common, many being arranged to deal with all three operations.

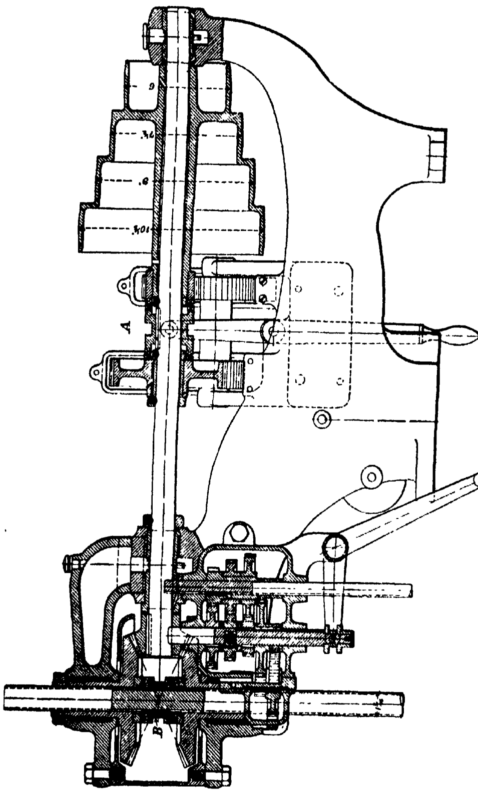


Fig. 101.—Driving arrangement of Pollard drill.

A typical driving arrangement for a *vertical drilling machine* is shown in Fig. 101, from the practice

f Messrs. F. Pollard & Co., Ltd., Leicester. The cone pulley may drive the shaft direct through the clutch at A, when this is on the right-hand side, or through double gear when the clutch is on the left-hand side. Thence through bevel gearing direct to the spindle, a clutch at B allowing for either top or bottom bevel to be engaged and so give a reverse motion for use when tapping. The other set of gears shown are for the feed motion.

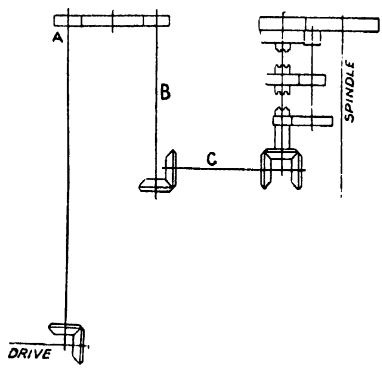


FIG. 102.—Diagram of radial drill drive.

Radial drilling machines and horizontal boring machines usually have a moving spindle head. The power from the line shaft or motor is then given out at one place on the machine, generally low down on either a base-plate or bed, and from this point transmitted to the spindle. In Fig. 102 is shown the diagram of the shafts and gearing used on the Cincinnati-Bickford *radial drill*. The drive from the first pair of bevel gears passes through the centre of the pillar of the machine, so that the whole of the

gears may swing round the driving gear A. The shaft at B allows the radial arm to be elevated, and the shaft at C is carried at the back of the radial arm, from which motion is transmitted to the spindle, first through a reversing motion, and then through three changes of gear.

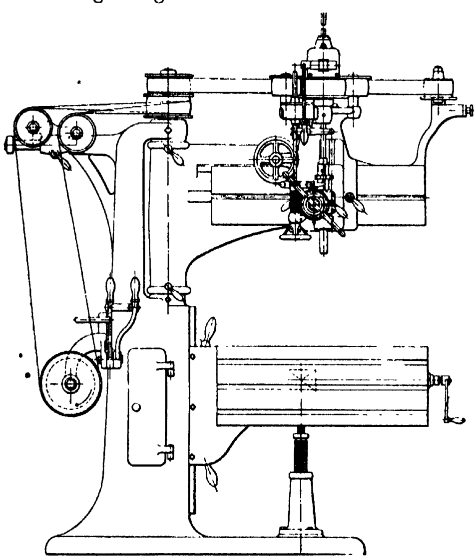


Fig. 103.—Example of belt driving.

Belts are frequently used instead of shafts. An example of a Pollard light radial drill so driven is given in Fig. 103. Whenever a belt pulley is used to drive a spindle direct, the pulley should not be carried on the spindle but on a bush supporting it, so that no bending action due to the pull of the belt

comes on the spindle. Two examples of belt pulleys so arranged, and illustrating the application of ball-bearings, are shown in Figs. 104 and 105.

In a *horizontal spindle boring machine* by Messrs. Geo. Richards & Co., Ltd., a vertical belt is used. Fig. 106 shows a front view of the arrangement, and

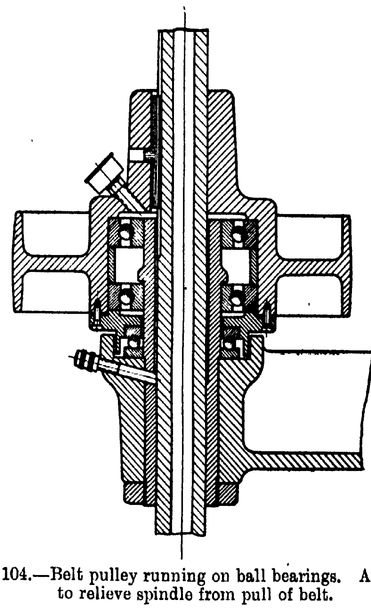


FIG. 104.—Belt pulley running on ball bearings. Arranged to relieve spindle from pull of belt.

Fig. 107 the run of the belt. From the driving gear box pulley D, the belt passes upwards to a tension pulley on top of the upright, then down to the head, round one jockey pulley, round a pulley A on the spindle, and back to the gear box. By this

method the head is allowed a vertical movement without altering the tension or effective driving length of the belt, and the belt has a large area of contact with the pulley A. This runs freely on the spindle which it drives through the gearing shown, first through the top shaft at two rates of speed, and then either direct to the boring head through the pinion

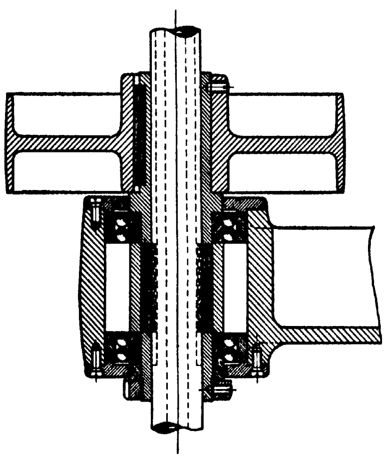


FIG. 105.—Belt pulley running on ball-bearings. Arranged to relieve spindle from pull of belt.

B at two rates of speed, or through the gears E and F. As there are eight changes of speed in the driving gear box, a total of thirty-two speeds is given to the spindle.

Right-angle Drive.—The arrangement of reversing gears shown in Fig. 108 allows the driving pulley to be placed on the same axis as the driving

shaft or at right angles to it. This is often a great convenience with a machine such as a boring machine, as when locating it in position in the shop it may be placed in the most convenient position for operating irrespective of the position of the line shaft.

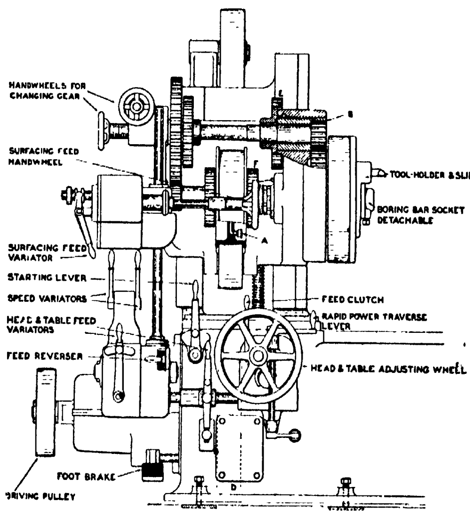


FIG. 106.—Richards' boring, facing, and milling machine.

A Vertical Spindle Milling Machine by the Becker Milling Machine Co., U.S.A. (Figs. 109 and 110), is a good example of direct belt-driving. A sectional view of the spindle head is given in Fig. 111. Note that the pulley is carried on a sleeve in a similar manner to Fig. 105. The arrangement of the gearing is clearly seen from Fig. 109. The pulley may

drive the spindle direct, the small handle at the top being used to lock it, or through one of the two sets of ordinary back gears, carried by eccentric studs so that they may be thrown in or out of gear as required. The large gear at the bottom and the one immediately above it are keyed to each other, and in turn to the spindle, this gear having a long sleeve passing through the pulley and having lock nuts A and B to keep the gear in

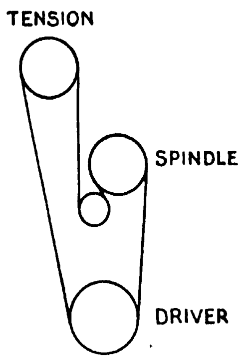


FIG. 107.

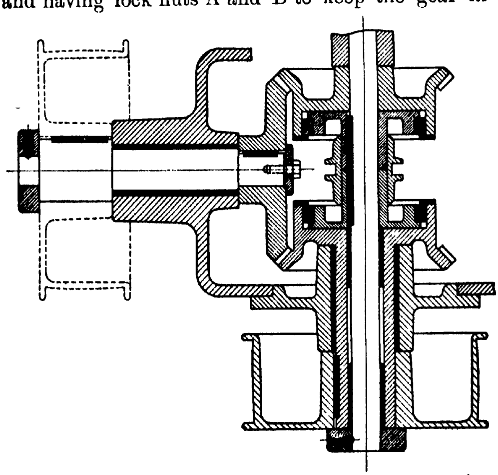


FIG. 108.—Arrangement for driving at right angles at option.

position (see Fig. 111). The gear to which the pulley is keyed has a sleeve and nut D.

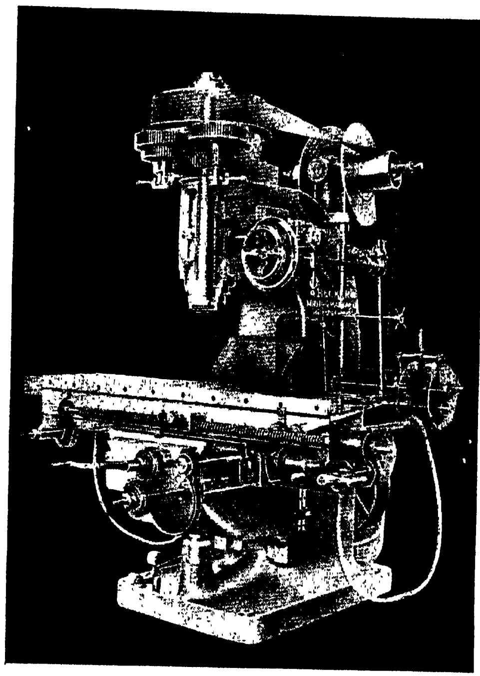


FIG. 109.—The Becker vertical milling machine.

The lower bearing of the spindle is carried in a conical sleeve L with adjustment for wear by the nuts F and H. Ball-thrust bearings J are provided

to take the end thrusts. The lock nuts at E keep the spindle in position relative to the sleeve.

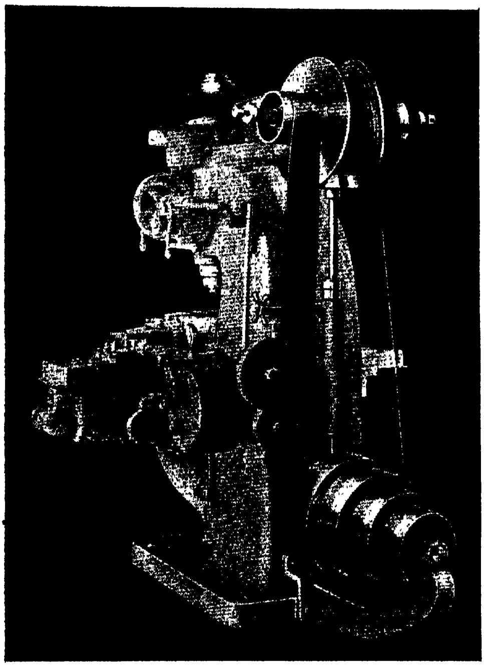


FIG. 110.—The Becker vertical milling machine.

The method of obtaining the feed motion from the driving belt is interesting. The rate of feed may be varied by fine steps by means of a cone rolling between the two belt pulleys at the top of the

machine. *Friction discs* are carried by these pulleys with tension springs so that they grip and drive the

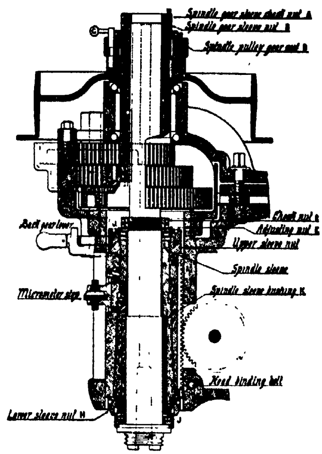


FIG. 111.

cone. This cone, carried on a sliding shaft, operates

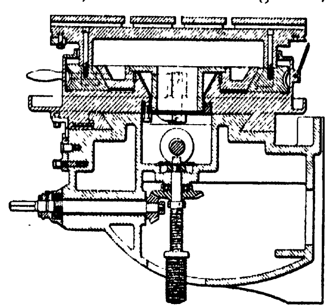


FIG. 112.—Section A-A through rotary table, table support and knee.

the various feeds, and may be raised or lowered

while in motion by rack and pinion control. Raising the cone brings it nearer to the centre line of the pulleys, and its motion is correspondingly diminished. Lowering it causes it to rotate faster as it ap-

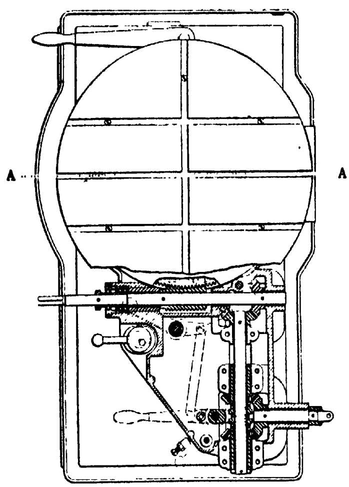


FIG. 113.—Rotary table driving mechanism.

proaches the outer edge of the pulley flanges. Figs. 112 and 113 show the construction and the driving arrangement of the *rotary table* for continuous milling, and is self-explanatory. Note the double ball thrust to the worm shaft.

CHAPTER VIII.

MISCELLANY.

Clutch for Power Press.—An ingenious arrangement of clutch and operating mechanism for a power press is made and patented by Messrs. John Hands & Sons, Ltd., Birmingham. The clutch is

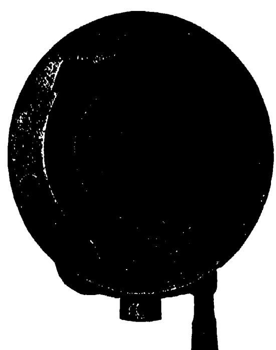


FIG. 114.—Patented clutch to give single or continuous revolution.

shown in Figs. 114 to 116, and a press to which it is fitted in Fig. 117. It is a combination clutch, and is such that upon the depression of a treadle or other lever, a positive one revolution only can be obtained whether the treadle is retained or let go.

Or it may be set for continuous running and stopping at will.

The construction and action of the clutch can be clearly followed from Fig. 114, showing it in running position, and Fig. 115 showing it at rest. The direction of rotation is anti-clockwise.

A is a steel driven block keyed to the crankshaft B, and upon this block the swinging driving pawl D is carried by a hinged joint, so that its

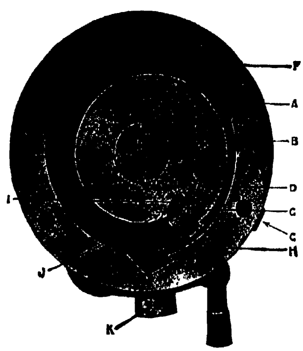


FIG. 115.—Patented clutch to give single or continuous revolution.

free or forward end is pushed outwards by the spring E, to engage with recesses in a steel driving ring F, which is securely bolted to the driving gear. Fixed at the side of pawl D is a wing or catch-piece G, which on the completion of the stroke comes into contact with a cam lever H, pivoted to the outer casing, thereby forcing the driving pawl D out of the recess in the driving ring, which continues to run freely, but allows the crankshaft B to come to a standstill.

Fixed to the side of the press is a stationary disc, surrounding which is a partly rotatable plate, and upon this latter the cam lever H is carried. This cam lever is suitably connected to the treadle motion. Thus far, when the cam lever is withdrawn and retained, the crankshaft B is driven continuously, but when let go the shaft is stopped when the pawl reaches the cam lever, and remains so.

FIG. 116.—Patented clutch to give single or continuous revolution.

A screw with a tommy head is provided whereby the cam lever H may be fixed to the partly rotating plate above mentioned, and also the same screw may be used to secure the plate to the stationary disc, leaving the cam lever H free to move upon its pivot. Assuming that the rotating plate is fixed and the cam lever free, the tendency of the latter is to be pulled inwardly by a strong spring contained inside

the stationary disc, whereby the pawl is lifted free

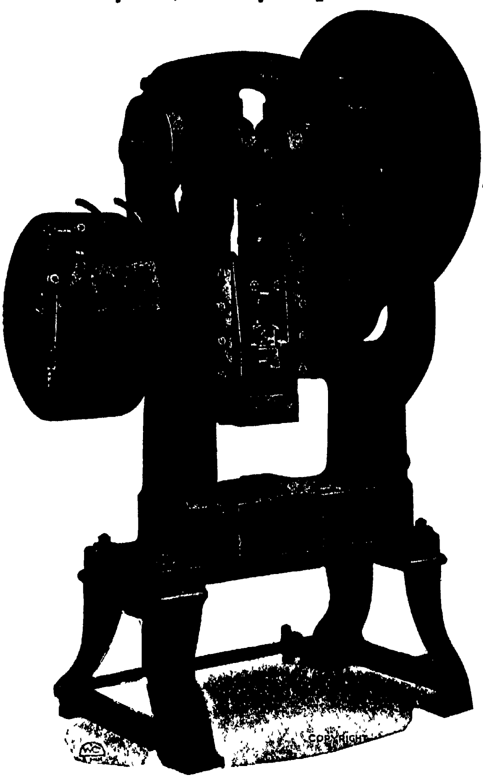


FIG. 117.—Power press fitted with patent clutch, made by Messrs. John Hands & Sons Ltd., Birmingham.

from the driving recess in plate F and the shaft thereby stopped. When thus stopped suddenly there

is a tendency from the momentum of the shafts and other parts of their being carried beyond a given point, but the projecting piece G on the pawl comes in contact with and is arrested by one of the two studs I, J, which are carried by the stationary disc, and thus give a definite stopping point. When the cam lever H is pulled outwards the pawl is free to follow, and to engage with the driving recess, and as long as thus held outwards the shaft is rotated.

Assuming now that the cam lever is fixed to the partly rotating plate and the latter is free to move through its limited distance, then by moving this plate backward (against its spring) the cam surface is turned clear of the pawl, when the latter will move outwardly, but in this case the cam lever was not moved radially out of the path of the pawl, consequently the pawl comes in contact therewith and is raised clear of the driving recess, and thus the shaft is stopped, even though the treadle may not have been liberated; but if the treadle has been liberated, the partly rotating plate is merely moved forward to its normal position, in which the cam plate still lies in the path of the pawl. If arrested as first stated the catch on the pawl engages with the first stud, and if arrested as secondly stated the catch engages with the second stud, in each case to give a positive point of cessation of movement.

Differential Epicyclic Gearing.—A very large reduction ratio is possible by the use of differential gearing. This may be arranged in two forms, as shown in Figs. 118 and 119, and consists essentially of a fixed gear A, a planetary gear B, and a driven gear D keyed to the shaft S. They may be enclosed

in a pulley as shown, thus making a very compact and dust-proof arrangement and allowing the gears to run in oil.

Referring to Fig. 118 the differential pinion B has a wide face and meshes with both gears A and D. Assuming A to have 30 teeth and D 29 teeth, if the driving pulley makes exactly one revolution, pinion B

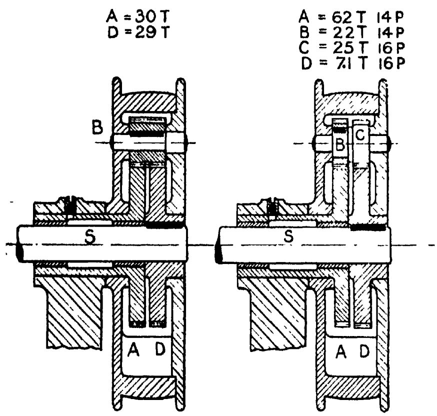


FIG. 118. FIG. 119.
Differential epicyclic gearing.

is compelled to engage every one of the 30 teeth on the fixed gear A; at the same time the teeth of the pinion B must come into engagement with 30 teeth on gear D, and as gear D has only 29 teeth, it is consequently compelled to move in a direction *opposite* to the rotation of the pulley to make good for the difference of one tooth. Thus, for every revolution of the pulley, D is thrown back one tooth, and it will take 29 revolutions of the pulley to produce one turn

of the shaft. The reduction of this particular train is therefore 29 to 1.

If the numbers of teeth of A and D were *reversed*, that is, fixed gear A had 29 teeth and D 30 teeth, for 1 revolution of the pulley gear D would be compelled to advance one tooth, and the direction of rotation of the shaft would then be in the *same direction* as that of the driving pulley.

By having large numbers of teeth in the gears very large ratios may be obtained, but this can be effected with comparatively small gears by introducing different pitches for the gears, and instead of the wide-faced pinion B meshing with both gears A and D, making this as two separate gears, fixed to each other, or four gears in all instead of three. This is shown in Fig. 119, and besides the large reduction possible has the advantage that the gears are properly in mesh, which of course is not so with the three-gear system.

A typical example is as follows: gear A has 62 teeth, 14 P.; gear B has 22 teeth, 14 P.; gear C 25 teeth, 16 P., and gear D has 71 teeth, 16 P. It will be observed that these teeth give the correct centre distances for each pair.

To find the ratio of the train we have

$$\frac{62}{22} \times 25 = 70{\cdot}454.$$

This is the number of teeth on gear D that C passes in one complete revolution of the driving pulley. Since gear D has 71 teeth, it is compelled to move through the difference, or

$$71 - 70{\cdot}454 = 0{\cdot}546 \text{ and } \frac{71}{{\cdot}546} = 130 \text{ to } 1.$$

This type of differential gear is very frequently used to give slow and quick motion to a feed shaft, a clutch being provided so that either the pulley may

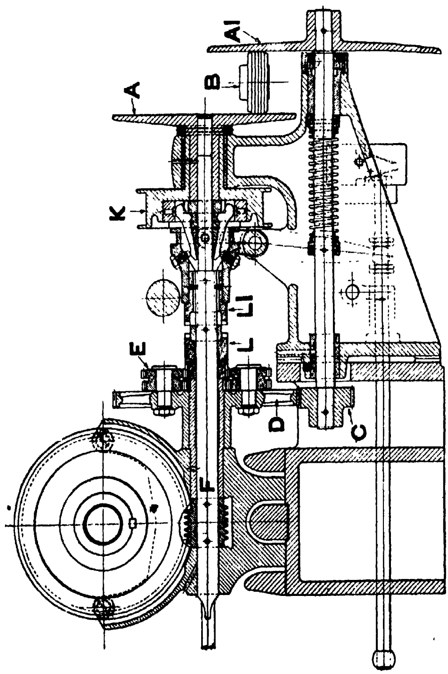


FIG. 120.—Mechanism of Cleveland automatic screw machine.

be clutched direct to the shaft or the gear D to the shaft.

An application of the differential gear is shown in Fig. 120, being a portion of the mechanism of the Cleveland Automatic Screw Machine in which is

embodied a slow speed at a varying rate, and a fast speed at a constant rate. The power is transmitted

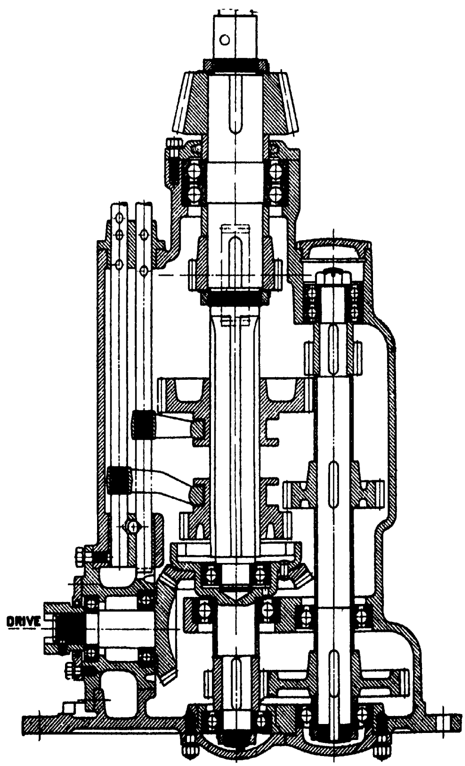


FIG. 121.—Example of ball-bearing application.

through the pulley K, running freely on shaft F. The pulley is attached to the friction disc A, which,

through the bowl B drives the disc A1 at varying speeds, depending upon the position of bowl B, which may be moved to or from the centres of the discs. Thence the motion is transmitted through the gears C and D. The latter carries a double planetary pinion E, one portion of which rotates around a stationary gear, and the other portion drives a gear that is part of a clutch L at a reduced speed

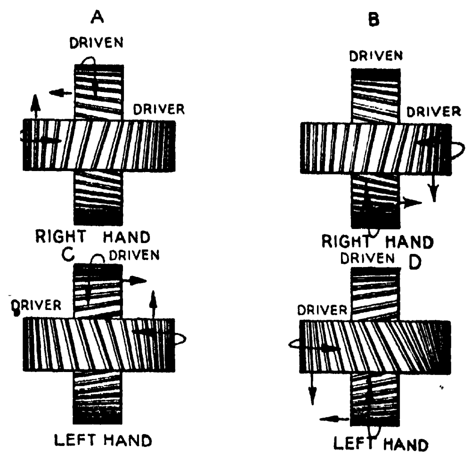


FIG. 122.—Direction of rotation and thrust of spiral gears.

due to the differential motion of the planetary gears. The clutch L1 is keyed to shaft F and is slidable on it, and according to its position engages with clutch L through teeth, so that the shaft is driven at a slow motion, or with the friction clutch (expanding-ring type) in the pulley K, as shown. The shaft then runs at a fast speed without any reduction.

Ball-bearing Application.—Fig. 121 shows an

interesting example of right-angle driving. There are many details in this illustration which will well repay study, including the application of ball-bearings, and the use of the castellated shaft for the sliding gears.

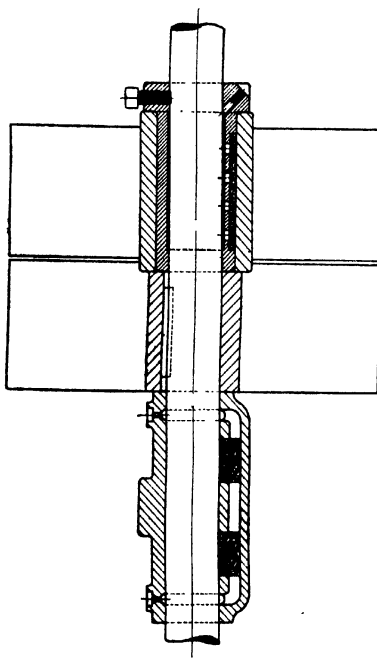


FIG. 123.—Fast and loose pulleys for counter-shaft.

Direction of Rotation of Spiral Gears.—The diagrams in Fig. 122 will be helpful in determining the direction in which spiral gears will rotate, and the side that takes the thrust.

Counter-shafts.—The counter-shaft of a machine tool being generally placed overhead is often the subject of neglect. For this reason, care should be

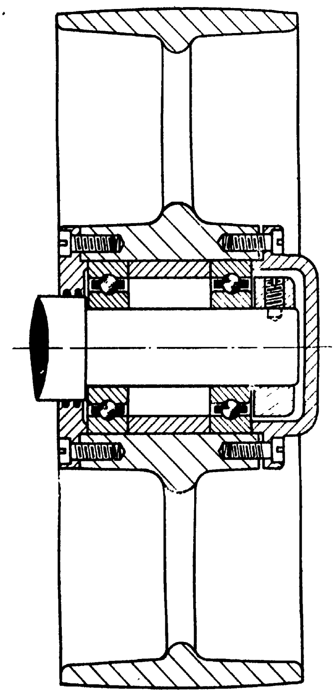


FIG. 124.—Ball-bearing loose pulley.

taken in the design that there is an ample reserve of lubrication for all running parts. The loose pulley should run on a bush preferably provided with a good oil chamber and a pad of felt to distribute the oil

evenly on the running surface (see Fig. 123). The hanger bearing should have a good oil well from which the oil may be circulated through the bearing and back to the oil well. This may be either by wick or oil rings.

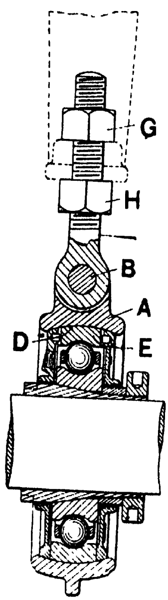


Fig. 125.

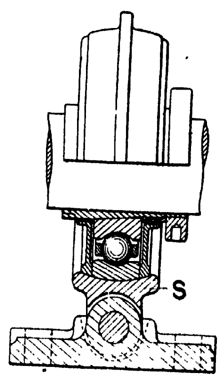


Fig. 126.

Ball-bearings play a useful part with countershafts as they will run for a long period with one supply of oil, provided they are carefully mounted and enclosed that the oil may be retained. They may be used for both pulleys (as Fig. 124) and bear-

ings. Figs. 125-7 show the Hoffman shaft bearing. The housing A, of cast iron, is pivotally hung on a stud B suspended by eye-bolts C at three points, and capable of being adjusted for height and locked in position by the nuts GH. The housing has a spherical

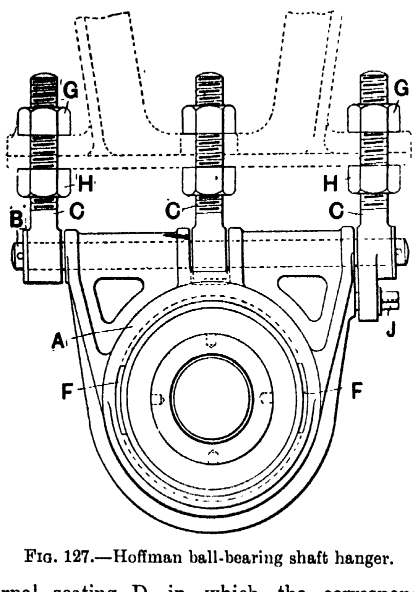


FIG. 127.—Hoffman ball-bearing shaft hanger.

internal seating D, in which the corresponding spherical face of the ring E of the bearing is capable of rocking. This bearing is inserted in its housing by turning the two rings at right angles to the housing, and passing them into it through the gaps FF which are provided for the purpose. It will be seen

that the whole of the bearing can swing on the stud B so that side thrust is prevented from coming upon the ball races, and that irregularity of alignment is allowed for by the spherical seat only. To prevent the whole shaft from swinging, one of the bearings is provided with a lock screw J. In Fig. 126 the housing S is arranged for carrying as a pedestal.

Friction Clutch Counter-shaft.—Friction clutches are used on counter-shafts instead of fast and loose pulleys, but this is generally to be found in American practice, British makers preferring the former as giving least trouble. Generally a cone clutch is used, operated by small bell-crank levers. Fig. 128 shows an example by the Smith Counter-shaft Co., U.S.A. Only one belt is used to drive it, and a feature of the design is a differential gear which enables it to give two forward and one reverse speeds. There are three principal parts, viz. a pulley A with a spur gear B attached to the hub, an internal gear fitted with frictions, F_3 and F_4, and a three-armed spider E carrying the intermediate pinions which mesh between the internal and the spur gear, and fitted with frictions F_1 and F_2. The functions of the four frictions are as follows: When engaged, friction F_1 clutches the spider E to the shaft; friction F_2, when engaged, holds the spider stationary and prevents it from rotating; friction F_4, when engaged, clutches the internal gear to the shaft. The outer members of frictions F_1 and F_4 are keyed to the shaft, while the outer members of frictions F_2 and F_3 are anchored to the overhead timbers or other convenient stationary members. There are two shipper rods required, operated if desired by one shipper lever. When

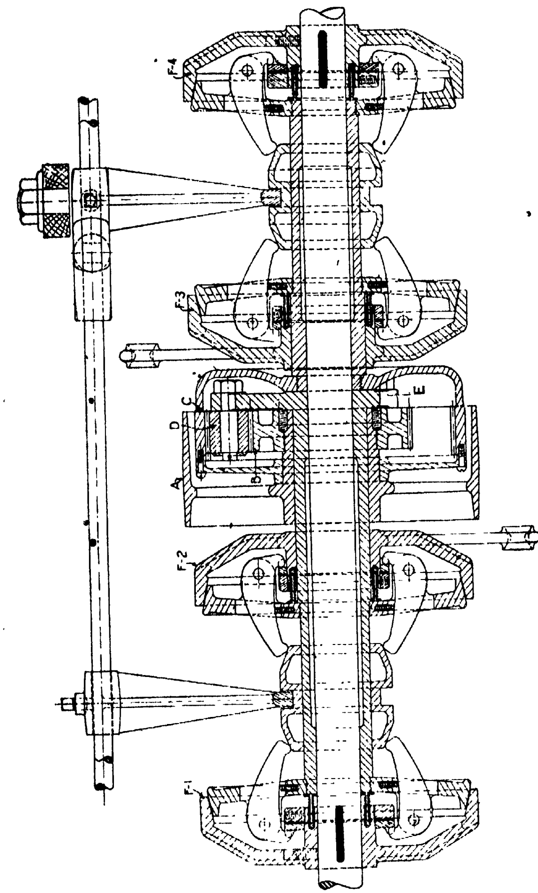


Fig. 128.—One-belt two-speed and reversing counter-shaft.

frictions F_1 and F_3 become engaged, they give th slow speed forward, which is a powerful geared driv the 13-in. pulley doing the work of a 24-in. on With the spools in this position, the internal gear is held stationary, and the three-armed spider fro which the gear drive is taken is clutched to the shaf

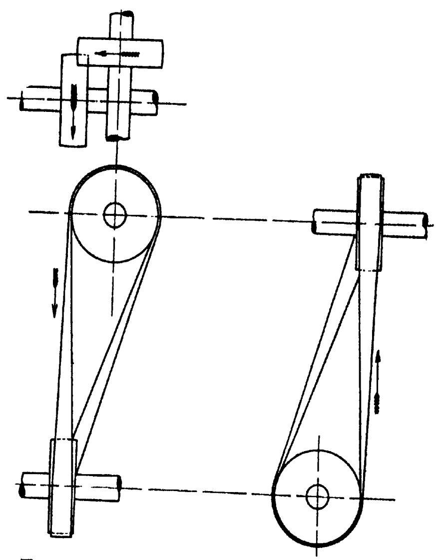


FIG. 129.—Direction of rotation of half-twist belt.

By throwing the shipper rod over to extreme right frictions F_2 and F_4 become engaged, giving the reverse drive. The spider E is then held stationary by friction F_2, and the internal gear, which is then reversed, is clutched to the shaft.

The high-speed forward movement is obtained by

making a positive connection between the pulley and the shaft, driving it with and at the same speed as the pulley. The counter-shaft may be stopped or started by disengaging or engaging friction F_4.

Direction of Rotation with Half-twist Belt.—With half-twist belt driving, it must always be borne in mind that the belt must *run on to a pulley in a straight line* (see Fig. 129). It may pull off at an angle.

Rules for Calculating the Speed of Pulleys and Gears.

In any system of pulleys or gears the general rule holds that the product of the diameters or number of teeth of the driving wheels and the number of revolutions per minute of the first driver must equal the product of the diameters or number of teeth of the driven wheels and the number of revolutions per minute of the last driven wheel.

The most frequent pulley calculations relate to the speeds of machines and counter-shafts, for which we have the four following rules, based upon the above principle.

Rule I.—Revolutions of driven

$$= \frac{\text{Diam. of driver} \times \text{revs. of driver}}{\text{Diam. of driven}}.$$

Rule II.—Diameter of driven

$$= \frac{\text{Diam. of driver} \times \text{revs. of driver}}{\text{Revs. of driven}}.$$

Rule III.—Revolutions of driver

$$= \frac{\text{Diam. of driven} \times \text{revs. of driven}}{\text{Diam. of driver}}.$$

Rule IV.—Diameter of driver

$$= \frac{\text{Diam. of driven} \times \text{revs. of driven}}{\text{Revs. of driver}}.$$

To find the speed when three or more shaft are connected by belts :—

Rule.—Multiply together the number of revolutions per minute of the first driver, and the diameters of each driver, and divide by the product of the diameters of the driven pulleys. The quotient will be the number of revolutions per minute of the last driven.

To find the diameter of a pair of gears when the centre distance and ratio are fixed :—

Rule.—As the sum of the ratio is to one of them, so is the distance apart of centres to the radius of the other one.

Example.—Ratio of speeds 5 to 3. Distance between centres 12 ins.

$$\text{as } 5 + 3 : 5 : 12 : : R$$

$$R = \frac{12 \times 5}{8} = 7\tfrac{1}{2} \text{ ins., and } 12 - 7\tfrac{1}{2}$$

$$= 4\tfrac{1}{2} \text{ ins.} = \text{radius of the smaller.}$$

APPENDIX.

Horse-Power that Different Leather Belts will Transmit per Inch in Width at Various Speeds.

Velocity of Belt per minute.	Kind of Belts.								
	Best Oak-tanned Belt.			Best Link or Chain Belts.					
	Single Belts.	Light double Belts.	Heavy double Belts.	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1
Feet.	Horse-Power they will Transmit.								
100	·15	·21	·27	·13	·15	·17	·20	·24	·27
200	·30	·42	·55	·25	·29	·35	·40	·47	·55
300	·45	·64	·82	·35	·44	·52	·60	·71	·82
400	·61	·85	1·09	·51	·58	·69	·80	·95	1·09
500	·76	1·06	1·36	·64	·73	·86	1·00	1·18	1·36
600	·91	1·27	1·64	·76	·87	1·04	1·20	1·42	1·64
700	1·06	1·49	1·91	·89	1·02	1·21	1·40	1·65	1·91
800	1·21	1·70	2·18	·92	1·16	1·38	1·60	1·89	2·18
900	1·36	1·91	2·45	1·05	1·31	1·55	1·80	2·13	2·45
1000	1·51	2·12	2·73	1·27	1·45	1·73	2·00	2·36	2·73
1100	1·67	2·33	3·00	1·40	1·60	1·90	2·20	2·60	3·00
1200	1·82	2·55	3·27	1·53	1·75	2·07	2·40	2·84	3·27
1300	1·97	2·76	3·55	1·65	1·89	2·25	2·60	3·07	3·55
1400	2·12	2·97	3·82	1·78	2·04	2·42	2·80	3·31	3·82
1500	2·27	3·18	4·09	1·91	2·18	2·59	3·00	3·55	4·09
1600	2·42	3·39	4·36	2·04	2·33	2·76	3·20	3·78	4·36
1700	2·58	3·61	4·64	2·16	2·47	2·94	3·40	4·02	4·64
1800	2·73	3·82	4·91	2·29	2·62	3·11	3·60	4·25	4·91
1900	2·88	4·03	5·18	2·42	2·76	3·28	3·80	4·49	5·18
2000	3·03	4·24	5·45	2·55	2·91	3·45	4·00	4·73	5·45
2100	3·18	4·45	5·73	2·67	3·05	3·63	4·20	4·96	5·73
2200	3·33	4·67	6·00	2·80	3·20	3·80	4·40	5·20	6·00
2300	3·49	4·88	6·27	2·93	3·35	3·97	4·60	5·44	6·27
2400	3·64	5·09	6·55	3·05	3·49	4·15	4·80	5·67	6·55
2500	3·79	5·30	6·82	3·18	3·64	4·32	5·00	5·91	6·82
2600	3·94	5·52	7·09	3·24	3·78	4·49	5·20	6·15	7·09
2700	4·09	5·73	7·36	3·28	3·85	4·66	5·40	6·38	7·36
2800	4·24	5·94	7·64	3·31	3·86	4·73	5·60	6·62	7·64
2900	4·39	6·15	7·91	3·32	3·87	4·78	5·80	6·85	7·91
3000	4·50	6·36	8·18	3·31	3·86	4·75	5·97	7·09	8·18
3100	4·60	6·58	8·45	3·30	3·85	4·73	5·96	7·33	8·45
3200	4·69	6·79	8·70	3·28	3·82	4·71	5·94	7·37	8·73
3300	4·77	7·00	8·86	3·24	3·77	4·70	5·92	7·35	8·88
3400	4·84	7·21	8·90	3·19	3·71	4·64	5·87	7·32	8·86
3500	4·90	7·31	9·06	3·13	3·61	4·50	5·78	7·26	8·80
3600	4·95	7·40	9·16	3·05	3·50	4·37	5·67	7·16	8·78
3700	4·99	7·48	9·24	2·96	3·39	4·26	5·55	7·01	8·58
3800	5·03	7·54	9·29	2·84	3·28	4·15	5·41	6·87	8·41
3900	5·06	7·60	9·34	2·72	3·13	4·02	5·20	6·70	8·27
4000	5·08	7·64	9·37	2·58	2·95	3·84	5·01	6·48	8·04

Surface Speeds of Various Diameters and Revolution

Feet per Min. / Diam. Inch.	10.	12.	14.	16.	18.	20.	25.	27½.	30.	35
	Turns per Minute.									
1/4	153	188	214	244	275	306	382	420	458	535
3/8	102	122	143	163	183	204	255	280	306	357
1/2	76	92	107	122	137	153	191	210	228	267
5/8	61	73	86	98	110	122	153	168	183	214
3/4	51	61	71	81	92	102	127	140	153	178
7/8	44	52	61	70	79	87	109	120	131	153
1	38	46	53	61	69	76	95	105	114	134
1 1/8	34	41	47	54	61	68	85	93	102	119
1 1/4	31	37	43	49	55	61	76	84	91	107
1 3/8	28	33	39	44	50	55	69	76	83	97
1 1/2	25	30	36	41	46	51	64	70	76	89
1 3/4	22	26	30	35	39	44	55	60	65	76
2	19	23	27	30	34	38	48	52	57	67
2 1/4	17	20	24	27	30	34	42	47	51	59
2 1/2	15·27	18·3	21	24	27	31	38	42	46	53
2 3/4	13·8	16·6	19·4	22	25	28	35	38	42	49
3	12·7	15·2	17·8	20	23	25	32	35	38	44
3 1/4	11·7	14·1	16·4	18·8	21	23	29	32	35	41
3 1/2	10·9	13·1	15·3	17·4	19·6	22	27	30	33	38
3 3/4	10·18	12·2	14·2	16·3	18·3	20·3	25	28	30	36
4	9·5	11·4	13·4	15·2	17·2	19·1	24	26	29	33
4 1/2	8·5	10·2	11·9	13·6	15·3	17·0	21	23	25	30
5	7·6	9·2	10·7	12·2	13·7	15·27	19·1	21	23	27
5 1/2	6·9	8·3	9·7	11·1	12·5	13·9	17·4	19·1	20·8	24
6	6·4	7·6	8·9	10·2	11·5	12·7	15·9	17·5	19·1	22
6 1/2	5·85	7·0	8·2	9·4	10·6	11·7	14·7	16·1	17·6	20·5
7	5·45	6·5	7·6	8·7	9·8	10·9	13·6	15·0	16·4	19·1
7 1/2	5·1	6·1	7·1	8·1	9·2	10·2	12·7	14·0	15·3	17·8
8	4·75	5·7	6·7	7·6	8·6	9·5	11·9	13·1	14·3	16·7
8 1/2	4·5	5·4	6·3	7·2	8·1	9·0	11·2	12·3	13·5	15·7
9	4·25	5·1	5·9	6·8	7·6	8·5	10·6	11·7	12·7	14·9
9 1/2	4·0	4·8	5·6	6·4	7·2	8·0	10·0	11·0	12·0	14·0
10	3·8	4·6	5·3	6·1	6·9	7·6	9·5	10·5	11·5	13·36
11	3·45	4·16	4·8	5·5	6·27	6·9	8·7	9·5	10·4	12·2
12	3·2	3·8	4·4	5·1	5·7	6·4	7·9	8·7	9·5	11·16
13	2·9	3·5	4·1	4·7	5·3	5·9	7·3	8·1	8·8	10·3
14	2·7	3·3	3·8	4·4	4·9	5·4	6·8	7·5	8·2	9·5
15	2·55	3·0	3·6	4·0	4·6	5·1	6·4	7·0	7·6	8·9
16	2·37	2·9	3·3	3·8	4·3	4·8	6	6·6	7·16	8·3
17	2·25	2·7	3·1	3·6	4	4·5	5·6	6·2	6·7	7·9
18	2·12	2·5	2·97	3·4	3·8	4·2	5·3	5·8	6·3	7·4
19	2·0	2·4	2·8	3·2	3·6	4·0	5·0	5·5	6·0	7·0
20	1·9	2·3	2·67	3·0	3·4	3·8	4·8	5·2	5·7	6·68
21	1·8	2·18	2·55	2·9	3·3	3·6	4·5	5·0	5·4	6·3
22	1·72	2·08	2·4	2·8	3·1	3·4	4·3	4·7	5·2	6·1
23	1·66	2·0	2·3	2·6	3·0	3·3	4·1	4·5	5·0	5·8
24	1·59	1·9	2·2	2·5	2·8	3·2	4	4·3	4·7	5·5

Surface Speeds of Various Diameters and Revolutions (*continued*).

Feet per Min.	40.	45.	50.	60.	70.	80.	90.	100.	110.	120.
Diam. Inch.	Turns per Minute.									
$\frac{1}{4}$	611	688	764	917	1070	1222	1375	1528	1681	1833
$\frac{3}{8}$	408	458	509	611	713	815	916	1018	1120	1222
$\frac{1}{2}$	306	344	382	458	535	611	688	764	840	916
$\frac{5}{8}$	244	275	306	367	428	489	550	611	672	733
$\frac{3}{4}$	204	229	254	306	357	407	458	509	560	611
$\frac{7}{8}$	175	196	218	262	306	349	393	436	480	523
1	153	172	191	229	267	306	344	382	420	458
$1\frac{1}{8}$	136	153	170	204	238	272	305	339	374	407
$1\frac{1}{4}$	122	138	153	183	214	244	275	305	336	366
$1\frac{3}{8}$	111	125	139	166	194	222	249	277	305	332
$1\frac{1}{2}$	102	114	127	153	178	204	229	254	280	305
$1\frac{3}{4}$	87	98	109	131	153	175	196	218	240	262
2	76	86	95	114	133	153	172	191	210	229
$2\frac{1}{4}$	68	76	85	102	119	136	153	170	187	204
$2\frac{1}{2}$	61	69	76	92	107	122	137	153	168	183
$2\frac{3}{4}$	55	62	69	83	97	111	125	139	152	166
3	51	57	64	76	89	102	115	127	140	153
$3\frac{1}{4}$	47	53	59	70	82	94	106	117	129	141
$3\frac{1}{2}$	44	49	54	65	76	87	98	109	120	131
$3\frac{3}{4}$	41	46	51	61	71	81	92	102	112	122
4	38	13	48	57	67	76	86	95	105	114
$4\frac{1}{2}$	34	38	12	51	59	68	76	85	93	102
5	30	31	38	46	53	61	69	76	84	92
$5\frac{1}{2}$	28	31	35	42	49	55	62	69	76	83
6	25	28	32	38	44	51	57	64	70	76
$6\frac{1}{2}$	23	26	29	35	41	47	53	59	65	70
7	22	25	27	33	38	44	49	51	60	65
$7\frac{1}{2}$	20·4	23	25	31	36	11	46	51	56	61
8	19·1	21	24	29	33	38	43	48	52	57
$8\frac{1}{2}$	18	20·2	22	27	31	36	10	45	49	54
9	17·0	19·1	21·2	25	30	34	38	42	47	51
$9\frac{1}{2}$	16·1	18·1	20·1	24	28	32	36	40	44	48
10	15·27	17·2	19·1	23	27	31	34	38	42	46
11	13·9	15·6	17·4	20·8	24	28	31	35	38	41
12	12·7	14·3	15·9	19·1	22	25	29	32	35	38
13	11·8	13·2	14·7	17·6	20·6	23	26	29	22	35
14	10·9	12·3	13·6	16·4	19·1	22	24	27	30	33
15	10·2	11·4	12·7	15·3	17·8	20·4	23	25	28	30
16	9·5	10·7	11·9	14·3	16·7	19·1	21·4	24	26	29
17	9·0	10·1	11·2	13·5	15·7	18·0	20·2	22	25	27
18	8·5	9·5	10·6	12·7	14·8	17·0	19·1	21	23	25
19	8·0	9·0	10·0	12·0	14·1	16·1	18·1	20	22	24
20	7·6	8·6	9·5	11·4	13·3	15·3	17·2	19	21	23
21	7·3	8·2	9·1	10·9	12·7	14·6	16·4	18	20	22
22	6·9	7·8	8·7	10·4	12·1	13·9	15·6	17·3	19	20·8
23	6·6	7·5	8·3	10·0	11·6	13·3	14·9	16·6	18·2	19·9
24	6·3	7·16	7·9	9·5	11·1	12·7	14·3	15·9	17·5	19·1

DIAMETRAL PITCHES FOR 15° INVOLUTE AND CYCLOIDA
CAST-IRON GEAR TEETH, PER INCH WIDTH OF FACE, FO
CONTINUOUS SERVICE IN ONE DIRECTION.—"MACHINERY.

Based on Wilfred Lewis' constants for speed and form c tooth. For ordinary steel castings, halve the pitch for co responding load, or double the load for corresponding pitch

Load in Pounds.	Diametral Pitch.							
	12-13 Teeth.	14-16 Teeth.	17-20 Teeth.	21-25 Teeth.	26-34 Teeth.	35-54 Teeth.	55-134 Teeth.	135 Teeth Rack
SPEED OF TEETH, 0 TO 100 FEET PER MINUTE.								
200	8	9	10	12	12	14	14	14
300	5½	6	7	8	8	9	9	10
400	4	4½	6	6	6	7	7	7
500	3¼	3½	4	4½	5	5	5½	6
600	3	3	3½	4	4	4¼	4½	5
700	2½	2½	3	3½	3½	3¾	4	4
800	2	2¼	2½	3	3	3¼	3½	3½
900	2	2	2¼	2½	2¾	3	3	3¼
1000	...	2	2	2¼	2½	2½	2¾	3
1100	...	...	2	2	2¼	2¼	2½	2¾
1200	...	...	...	2	2	2¼	2¼	2½
1300	...	...	...	...	2	2	2¼	2¼
1400	...	...	...	...	...	2	2	2
1500	..	...	...	...	...	...	2	2
SPEED OF TEETH, 100 TO 200 FEET PER MINUTE.								
100	12	14	16	...	...	...	...	...
200	6	7	8	9	9	10	11	12
300	4	4½	5	6	6	6	7	7
400	3	3½	4	4½	4½	5	5	5½
500	2½	2¾	3	3½	3½	4	4	4½
600	2	2¼	2½	2¾	3	3¼	3½	3¾
700	2	2	2¼	2½	2½	3	3	3¼
800	...	...	2	2¼	2¼	2½	2½	2¾
900	...	...	...	2	2	2¼	3¼	2½
1000	...	...	...	...	2	2	2	2¼
1100	...	...	...	...	...	...	2	2
1200	...	..	...	...	...	...	...	2

Diametral Pitches for 15° Involute and Cycloidal Cast-iron Gear Teeth, per Inch Width of Face, for Continuous Service in One Direction.—*Continued.*

Load in Pounds.	Diametral Pitch.							
	12-13 Teeth.	14-16 Teeth.	17-20 Teeth.	21-25 Teeth.	26-34 Teeth.	35-54 Teeth.	55-134 Teeth.	135 Teeth Rack
Speed of Teeth, 200 to 300 Feet per Minute.								
100	10	11	12	14	14	16	...	...
200	5	$5\frac{1}{2}$	6	7	7	8	8	9
300	$3\frac{1}{4}$	$3\frac{1}{2}$	4	$4\frac{1}{2}$	5	5	$5\frac{1}{2}$	6
400	$2\frac{1}{2}$	$2\frac{3}{4}$	3	$3\frac{1}{2}$	$3\frac{1}{2}$	4	4	$4\frac{1}{2}$
500	2	$2\frac{1}{4}$	$2\frac{1}{2}$	$2\frac{3}{4}$	3	3	$3\frac{1}{4}$	$3\frac{1}{2}$
600	...	...	2	$2\frac{1}{4}$	$2\frac{1}{2}$	$2\frac{1}{2}$	3	3
700	...	...	...	2	2	$2\frac{1}{4}$	$2\frac{1}{2}$	$2\frac{1}{2}$
800	...	...	...	...	2	2	2	$2\frac{1}{4}$
900	...	...	...	...	...	...	2	2
Speed of Teeth, 300 to 600 Feet per Minute.								
100	8	9	10	12	12	14	14	14
200	4	$4\frac{1}{2}$	5	6	6	7	7	7
300	$2\frac{1}{2}$	3	$3\frac{1}{4}$	$3\frac{3}{4}$	4	4	$4\frac{1}{2}$	5
400	2	$2\frac{1}{4}$	$2\frac{1}{2}$	3	3	$3\frac{1}{4}$	$3\frac{1}{2}$	$3\frac{3}{4}$
500	...	2	2	$2\frac{1}{4}$	$2\frac{1}{2}$	$2\frac{3}{4}$	3	3
600	...	..	...	2	2	$2\frac{1}{4}$	$2\frac{1}{4}$	$2\frac{1}{2}$
700	...	...	...	...	...	2	2	2
800	...	...	...	...	...	...	...	2
Speed of Teeth, 600 to 900 Feet per Minute.								
100	6	7	8	9	9	10	10	12
200	3	$3\frac{1}{2}$	$3\frac{3}{4}$	$4\frac{1}{2}$	$4\frac{1}{2}$	5	$5\frac{1}{4}$	$5\frac{1}{2}$
300	2	$2\frac{1}{4}$	$2\frac{1}{2}$	3	3	$3\frac{1}{4}$	$3\frac{1}{2}$	$3\frac{3}{4}$
400	...	...	2	$2\frac{1}{4}$	$2\frac{1}{4}$	$2\frac{1}{2}$	$2\frac{1}{2}$	3
500	...	...	...	...	2	2	2	$2\frac{1}{4}$

SPEED OF TEETH, 900 TO 1200 FEET PER MINUTE.

100	5½	5½	6	7	7	8	8	9
200	2	2¾	3	3½	3½	4	4	4½
300	...	2	2	2¼	2½	2½	2¾	3
400	...	...	...	...	2	2	2	2¼
500	...	...	...	...	...	...	...	2

SPEED OF TEETH, 1200 TO 1800 FEET PER MINUTE.

100	4	4½	5	5½	6	7	7	7
200	2	2¼	2½	2¾	3	3¼	3½	3¾
300	...	...	...	2	2	2¼	2½	2½
400	...	...	...	...	...	...	2	2

SPEED OF TEETH, 1800 TO 2400 FEET PER MINUTE.

100	3½	3¾	4¼	5	5	5½	6	6
200	2	2	2¼	2½	2½	2¾	3	3
300	...	...	...	...	...	2	2	2

TABLE OF CUTTING SPEEDS FOR VARIOUS METALS.

Cast iron . . . . 50 ft. per min.
Cast steel . . . . 60 " " "
Malleable iron . . . . . . 70 " " "
Machine steel forgings (15 to 20-point carbon) . . . . . . 65 " " "
Machine steel (black stock) . . 70 " " "
Tool steel forgings . . . . 35 to 40 " " "
Steel alloys (containing nickel and chromium) . . . . . 30 to 50 " " "
(Depending on alloy)
Yellow brass . . . . . 200 ft. per min.
Comp. brass . . . . 120 to 150 " " "
Bronzes . . . . . . 30 to 80 " " "
(Depending on alloy)

Sizes and Speeds of Motors to Drive Machine Tools.[1]

For variable speed, motors are recommended with a speed variation of 3 to 1.

For constant speed, motors are recommended with speeds varying from 1800 R.P.M. on small light machines, to 900 R.P.M. on heavy machines.

LATHES.

Swing, Ins.	Light Duty. Horse-power.	Medium Duty. Horse-power.	Heavy Duty. Horse-power.
14	2	3	5
16	3	5	5
18-20	3	5	7 : 5
22-24	5	7 : 5	10
27-30	$7\frac{1}{2}$	10	15
36-48	$7\frac{1}{2}$	10	20

Special Lathes.

Type.	Horse-power.
Car wheel, 48 in. . . .	20
Double axle, moderate duty	15
Heavy duty . . .	25

Driving Wheel Lathes.

Size, ns.	Horse-power.
51	15
60-69	20
79	25
84	25
90	30
100	50 5 tail stock.

Note.—The average load factor for motors driving lathes is from 10 to 25 per cent. On some special machines, as driving-wheel and car-wheel lathes, the cuts are all heavy, which increases the average load factor to from 30 to 40 per cent.

[1] From Journal, "American Society of Mechanical Engineers," 1910.

Boring and Turning Mills.

Size.	Horse-power.
24-30 ins.	5
36-42 „	7½
60-90 „	10 and 5 for cross slide
100 „	15 and 5 „ „ „
10 ft.	20 and 7½ „ „ „
14 „	25 and 7½ „ „ „
16 „	30 and 10 „ „ „

Planing Machines.

Medium Duty.		Heavy Duty.	
Size, Ins.	Horse-power.	Size, Ins.	Horse-power.
24 × 24	5	24 × 24	7½
30 × 30	7½	42 × 42	25
36 × 36	10	56 × 56	25
		Frog and Switch	30
48 × 48	15	12 ft. × 10 ft.	60
56 × 56	15	14 ft. × 12 ft.	75
		Elevating cross-slide	12

Shaping Machines.

Size, Ins.	Horse-power.
14-20	3
24	5
36	7½

Slotting Machines.

Size, Ins.	Horse-power.
10	3-5
10-16	5-7½
20	7½-10
26-30	15
24-60	20

Note.—The work done on shapers and slotters is of a varying character. With light work the load factor will not exceed from 15 to 20 per cent; with heavy work, the load factor will be as high as 40 per cent.

Drilling Machines.

Radial Drills.		Upright Drills.	
Size, Ft.	Horse-power.	Size, Ins.	Horse-power.
4	3	Friction	$\frac{1}{4}$
5	5	15	$\frac{1}{2}$
6	5	20-26	1
10	$7\frac{1}{2}$	28-34	2
		42-50	3

Multiple-Spindle Drills.

Size, Ins.	Horse-power.
4-2	$7\frac{1}{2}$
6-2	10
8-2	10

MILLING MACHINES.

Horizontal—Plain or Universal.

Table Feed, Ins.	Cross Feed, Ins.	Vertical Feed, Ins.	Horse-power Mod. Heavy.
24	8	18	3
30	10	18	5-$7\frac{1}{2}$
36	12	20	$7\frac{1}{2}$-10
50	12	20	10-15

Vertical Milling Machines.

Table Diameter, Ins.	Spindle Diameter, Ins.	Horse-power.
28	4	5
32	4	$7\frac{1}{2}$
40	$4\frac{1}{2}$	10
54	5	15
70	6	20

Slab Milling Machines.

Width of Table, Ins.	Horse-power.
24-30	10
36	15
60	25
36 heavy	25
42 heavy	50

Note.—For the average milling operations the load factor averages from 10 to 25 per cent. On slab milling-machines where large quantities of metal are removed it will average from 30 to 40 per cent. The work on this class of machinery is usually light, and much time is required in making adjustments. Hence the load factor is rarely higher than 20 per cent.

Horizontal Boring, Drilling and Milling Machines.

Spindle, Ins.	Horse-power.
$3\frac{1}{2}$	3
4	5
5	$7\frac{1}{2}$
6	10
7	15

Cold Saws.

Diameter, Ins.	Thickness, Ins.	Horse-power.
12	$\frac{5}{32}$	2
15	$\frac{5}{32}$	2
18	$\frac{3}{16}$	3
20	$\frac{3}{16}$	3
24	$\frac{3}{16}$	5
32	$\frac{3}{16}$	$7\frac{1}{2}$
36	[illegible]	10

Horse-power Required for Grinding Machines.
The Churchill Machine Tool Co., Ltd.

Type of Machine.	Size of Machine.	Wheel.	Motor Required for Moderate Work.	Motor Recommended for Heaviest Work.
Plain grinders	4 in. model A	12 × 1½ ins.	3½ H.P.	5 H.P.
,, ,,	6 ,, ,, B	14 × 2 ,,	5 ,,	7½ ,,
,, ,,	10 ,, ,, B	16 × 2 ,,	5 ,,	10 ,,
,, ,,	12 ,, ,, C	18 × 3 ,,	10 ,,	15 ,,
,, ,,	14 ,, ,, D	20 × 3 ,,	12-15 ,,	20 ,,
,, ,,	18 ,, ,, E	26 × 3 ,,	15 ,,	25 ,,
,, ,,	22 ,, ,, F	26 × 3 ,,	15 ,,	30 ,,
Crankshaft grinder	—	24 ins.	12-15 ,,	20 ,,
Grinders with wheels up to	—	32 in. diameter	20 ,,	35 ,,
Universal grinders	10 ins.	12 × 1 ins.	3½ ,,	5 ,,
,, ,,	12 ,,	12 × 1½ ,,	3½ ,,	5 ,,
,, ,,	20 ,,	16 × 2 ,,	7½ ,,	10 ,,
Internal grinders	10 ,,	—	2½ ,,	3½ ,,
,, ,,	12 ,,	—	3½ ,,	5 ,,
,, ,,	16 ,,	—	3½ ,,	5 ,,
,, ,,	20 ,,	—	3½ ,,	5 ,,
Ring grinders	12 ,,	12 × ¾ ins.	3½ ,,	5 ,,
,, ,,	24 ,,	16 × 1½ ,,	5 ,,	7½ ,,
Vertical ring grinder	16 ,,	12 ins.	10 ,,	15 ,,
Piston ring grinders	30 and 40 ins.	16 ,,	7½ ,,	10 ,,
Vertical surface grinders	40, 60 and 72 ins.	14 ,,	15 ,,	30 ,,
Wet tool grinders	—	20 × 2 ins.	3 ,,	—

INDEX.

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